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About Authors

● C. R. Maxon *has lived with die castings for many years. Since joining the New Jersey Zinc Co. in 1923 he has spent most of his time informing both die casters and users of die castings of the research and development work carried on by his company. He has been a member of the Non-Ferrous Metals Division of the SAE Standards Committee since 1931. Engineers in many of the die casting plants, as well as motor car designers, Mr. Maxon states, furnished much helpful information for his paper.*

● Arthur Nutt *has been in charge of design and development of several firsts in the aircraft-engine industry. Among them are the Curtiss 600-700 hp. 12-cylinder chemically-cooled Challenger, the first engine to be designed and produced in quantity with this type of cooling, and the Curtiss 9-cylinder radial air-cooled R-1454 engine, the first of this type with a built-in supercharger and fully automatic lubricated valve gear. At the Wright Aeronautical Corp. many experimental engines have been engineered under his supervision, including Whirlwind and Cyclone engines up to 1500 hp. He was also in charge of design and development of the Curtiss D-12, Chieftain, Crusader and Challenger model engines and assisted in the development of Curtiss OX-5, OXX-6, V-2, K-12 and C-12 model engines. Mr. Nutt started his career as test engineer at Curtiss Aeroplane & Motor Co. immediately after receiving his B.S. degree in M.E. from Worcester Polytechnic Institute in 1916. Four years later he was made chief engineer. When Curtiss and Wright merged in 1930 he became vice-president in charge of engineering of the Wright Aeronautical Corp. Mr. Nutt has twice been aircraft-engine vice-president of the SAE, first in 1925 and again in 1930.*

● T. C. Smith *has been engaged in telephone work since receiving his B.S. in M.E. degree from Purdue University in 1910. Starting with the New York Telephone Co., his first work had to do with plant extension engineering and electric light and power engineering. In 1916 he*

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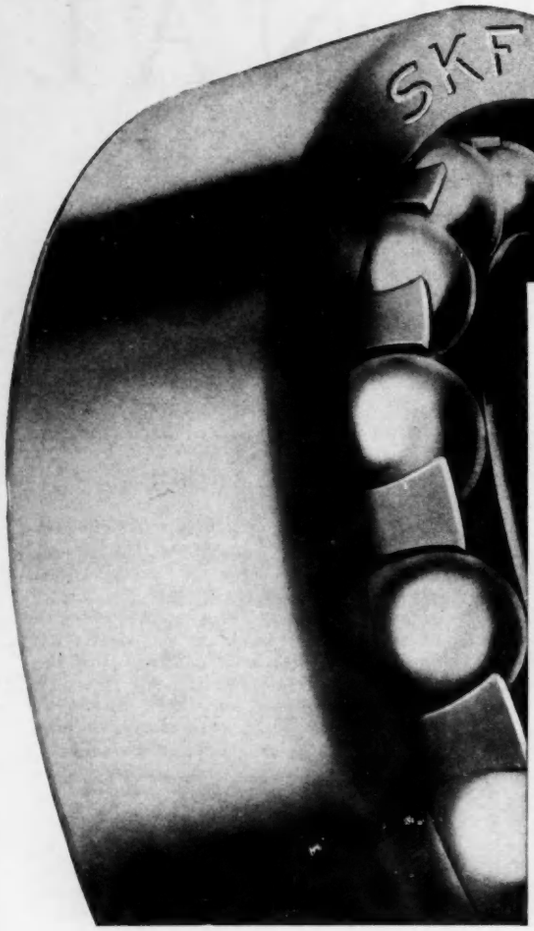

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BALL AND ROLLER BEARINGS

Utility Trucks, Cabs, Bodies and Auxiliary Equipment

By T. C. Smith
American Telephone & Telegraph Co.

BY means of representative cases this paper shows some of the marked developments in utility fleets in the past 41 years.

Among the examples discussed are a horse-powered truck of 1896, one of the first motor trucks used for the purpose in 1910, a number of light modern "mosquito-fleet" units adapted for special applications from light-weight passenger cars, heavier units with five- or seven-man cabs, and highly specialized units for erecting poles and pulling-in underground cables with their trailers and auxiliary equipment. Considerable technical information on the design and construction of these special bodies and their auxiliary equipment is presented.

In conclusion the author recommends four general objectives for consideration in rebuilding a fleet.

IN the utility field the transportation equipment performs a very important function in the operations of each company. It is responsible for maintaining constant touch with a widely distributed outside plant and with the subscribers' stations, any unit of which might require immediate attention at any time. The transportation equipment is called upon for necessary services in extending the plant as well as in the day-to-day maintenance. Also, in times of emergency, due to storms, floods, and other disasters the companies have become accustomed to lean heavily on the motor-vehicle fleets. The remarkable performances which have been turned in time after time prove that this confidence has not been misplaced.

I have observed that the motor-vehicle requirements in the electric and telephone companies are quite similar in many respects. The gangs, together with their materials and tools which in general are not dissimilar, must be moved at will on very short notice to any location where they may be required. On the job the truck serves as a headquarters for the gang; also, it supplies power which, through suitable power tools and apparatus, provides indispensable assistance to the gangs in performing their many heavy and difficult tasks.

[This paper was presented at the Regional Transportation and Maintenance Meeting of the Society, Baltimore, Md., April 15, 1937.]

The development of the characteristic utility fleet to its present state of advancement has required only a comparatively few years, but it represents in composite form the work of many able engineers and executives, both in the user companies and in the motor-vehicle manufacturing organizations. In this paper I do not, of course, wish to present a partisan viewpoint of any industry or organization, but rather the personal observations of one who has been closely associated with the development from its beginning. Naturally the examples which I have selected depict work in the telephone field, with which I am most familiar.

By means of a limited number of representative cases I shall show some of the marked developments along constructive lines as regards various items in the average utility fleet. However, this paper is not presented as a complete or finished picture. The utility fleets now cover too broad a field to discuss in complete detail in one paper. Also, judging from the new projects now in various stages of development, in the not-distant future we shall see advances in the fleets which will make our present equipment seem quite inadequate and unfinished.

Earlier Utility Transportation Equipment

By way of introducing the subject a few of the ancestors of the units which I shall discuss later are shown in Figs. 1-5. In them you will notice that there was a recognition of the need for many of the features which have been worked out in a fairly satisfactory and much more comprehensive manner in the latest present-day designs. To some of you these old pictures may suggest the days when any of us so bold as to visualize a completely motorized utility transportation department, were in danger of being considered impractical dreamers.

The light-duty construction truck shown in Fig. 1, it will be noted, carried side-box compartments on the body and had a combination rack for ladder and pike poles. The "Be Careful of Accidents" sign on the side-box indicates that our present safety movements are not of recent origin but were recognized as important factors in utility operation at least as far back as 1896. Also, it will be noted that the rear-right wheel of this truck carries a wire reel of which the foreman is quite proud since it was his invention.

The heavy-duty construction truck illustrated in Fig. 2 carried more materials, tools, and ladder equipment but afforded scant protection to the gang while moving or on the job.

Carefully prepared studies, about 1910, indicated the practicability and economy of utilizing gasoline-driven motor trucks for the transportation of men, supplies, and construction equipment of various kinds. The original studies, proving-in

scattered installations of truck equipment, were based upon using the trucks as purely transportation units, comparing them with horse-drawn vehicles. Lighting equipment is in evidence on these early trucks as shown in Fig. 3, but the design has not advanced as yet to the stage where windshields are required.

The general design of this truck, which was in service 25 years ago, would indicate that we do not have an entirely new conception in the present-day cab-over-engine units.

There was a marked tendency toward overloading light roadsters or "runabouts," as illustrated in Fig. 4, a common test of the limiting gross vehicle weight being to note whether the fenders actually touched the tires. This particular unit was used by an underground-cable-splicing foreman, as you will note by the manhole guard at the rear and the bilge pump at the side of the vehicle. I fear that present-day safety engineers might question the loading arrangement. However, there was no danger of this car operating at present-day speeds. This limitation was due not only to the design of the vehicle itself but also to the roads over which it was operated.

On a light construction truck there were, and still are, many small, as well as large, items of tools and material which had to be carried. In the particular instance shown in Fig. 5, three soap boxes were employed in addition to the large home-made box which is in evidence. Due to the inadequacy of body facilities, materials and tools were thrown together in the boxes as well as on the floor of the truck. By digging down into the large box you could locate various kinds of small material in the bottom where it was difficult to find. In getting down to it, sharp-edged tools, mixed loosely with the bulkier materials, had to be removed.

Lighter Vehicles in Utility Service

With this background let us jump ahead 20 or 25 years to the modern equipment. Now, it is not a matter of proving-in automotive over horse-drawn vehicles, but a question of obtaining proper automotive equipment.

There is a place in practically all utility fleets for many units of the lighter type mounted upon so-called "half-ton chassis" which are usually adaptations of light passenger cars. In a utility organization they are often referred to as the "mosquito fleet." These units ordinarily carry gross vehicle weights of 4000 to 4400 lb. with maximum limits ranging up to about 4800 lb. per unit. These weights would correspond with net loads ranging from 900 to 1800 lb. per car.

Drawers with flexible arrangements of removable partitions are provided together with slides, racks, and brackets for carrying larger equipment as shown in Fig. 6. There are also proper spaces for the smaller tools, as well as the permanent brackets for mounting the detachable wire reel shown on the outside of the body. Some idea of the amount of detail represented in the design of the body is obtained when it is realized that, in a body such as this, there may be incorporated as many as 175 drawer spaces, racks, tool-holders, and compartments.

Additional space inside the body across the front is accessible through a door on the right side just ahead of the reel. On the left side of the body are brackets for carrying various long tools, such as ladders, pike poles, and digging bars when these different items of equipment are needed.

In a body of this type practically every kind of material or tool which the workman ordinarily uses can be assigned to a properly located compartment as shown in Fig. 7. The arrangement should be such that all items are easily placed in and removed from the body and so that they will not be damaged in transportation. Materials or tools should be so arranged in the body that those which are associated with each particular kind of job are grouped together and so that those most frequently used are most readily accessible. In a body of this type all except the very long items of equipment are carried inside, protected from the weather. Such a body designed to meet the particular requirements of an operation can be produced at quite a low price considering the ample facilities provided.

Bodies of this kind can be designed to harmonize with the lines of the half-ton chassis and cabs so that they present a pleasing appearance as work units when seen on the highway. The outside of the body should, of course, be painted the same color as the cab and hood. For the inside it has been found that the use of aluminum paint considerably assists the vision of the workmen, particularly on dark days.

It will be noted that the roof on the body shown in Fig. 7 is made of a pressed-steel sheet rounded at the edges and corners and slightly raised toward the middle in order to drain properly and to present a pleasing appearance.

Where it may be necessary to carry some bulky material occasionally, such as brush from tree trimming, a number of insulators, or coils of wire, in addition to that already provided for in the body, a type of unit, somewhat similar to that mentioned above, can be used, with the provision of a flat top reinforced to carry the extra loads when required. Fig. 8 shows such a unit. It has strongly built sides and front rail around the top and also rope loops for use when required. In a top of this kind it is desirable to have a slightly raised rib across the rear in order to prevent material from sliding back. Also, it is necessary to provide for drainage of water which may be trapped on the top.

Information of a Technical Nature

It has been requested that this paper cover some of the technical data with regard to the materials used in the bodies, their dimensions, weights, and also the structural features which are incorporated in them. Bodies of the types just mentioned are ordinarily about 46 in. wide. Their height from floor to top outside is in the neighborhood of 33 in. In determining this dimension, of course, consideration must be given to the practicability of seeing back over the top of the body as viewed through the rear window from the rear vision mirror inside the cab at the top of the windshield. This provision assists in backing and is a very desirable safety feature, as well as a convenience that is much appreciated by the drivers. The lengths of these bodies vary from 5 to 6 ft.

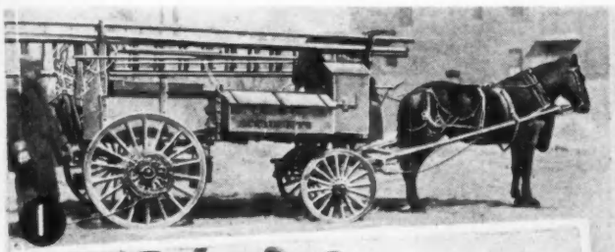
When made of steel, their weight, including internal drawers, partitions, and so on, ranges from about 500 to 650 lb., exclusive of the ladder brackets which add approximately 70 lb. to the weight. A body of this type made of aluminum alloy and 6 ft. long weighs about 325 lb.

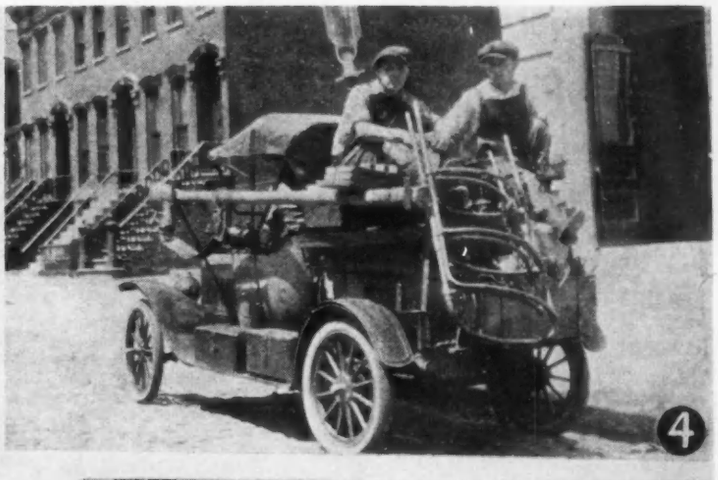
It seems that in public utility work the weight saving from using aluminum alloy is warranted in cases where its use permits keeping the gross vehicle weight within permissible limits for a particular size of chassis. Where the gross vehicle weight would remain within the desired limits using a steel body, it seems questionable whether the extra expense of the aluminum would be warranted. The low daily mileage has a major effect upon this matter. Utilities should, it seems, not be influenced too much by ton-mile considerations in studying the proper field for the much more expensive lighter-weight body materials.

"Galvannealed" steel is used quite generally for bodies. For those in the "mosquito fleet" 20-gage steel is employed for sides, roof, and floor; 24 gage for partitions, and 22 gage for drawers and fabricated doors.

The Galvannealed steel carries a thickness of about 0.002 in. of zinc on each side. This zinc coating is rolled while hot to prevent spangles and to present a smooth surface for painting.

The bodies are fabricated principally by electric spotwelding, using ⅛-in. to ¼-in. points. Some tack-welding is used and





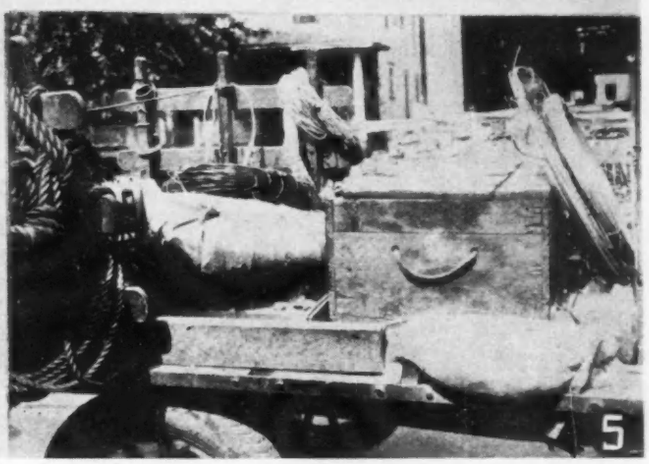


Fig. 1 – Light-Duty Construction Truck of More Than Forty Years Ago

Fig. 2 – Heavy-Duty Construction Truck of About the Same Period

Fig. 3 – Early Motor-Driven Truck used Purely as a Transportation Unit

Fig. 4 – Light Roadster Used by Underground-Cable-Splicing Crew

Fig. 5 – Typical Method of Carrying Tools and Materials on Early Light Construction Truck

practically no gas-welding. Less than 20 per cent of the joints are riveted.

The welding of Galvannealed steel requires more heat than does black sheet, and it is convenient to use water-cooled points in connection with the spotwelding. Timing of the welding operation is extremely important, and automatic control of timing and heat are necessary for quantity production in order to insure uniform results. A different technique of handling, therefore, is necessary in order to weld Galvannealed steel successfully. If practices ordinarily used with black sheet are attempted, the results are likely to be inferior and unsatisfactory.

In painting the Galvannealed surface, first it is cleaned with high-flash naphtha base cleaner. There are available primers and surfacers which have satisfactory holding qualities for paint on a Galvannealed surface. Because of their fairly good gloss without rubbing and good service in the field, present available synthetic enamels are favored for finishes of the all-metal utility bodies, except in some very special cases where these units are subjected to alkali dust, under which conditions lacquer finishes give better performance.

At the present time experimental work is being done in the use of phosphoric-acid treatment of the Galvannealed surface before painting in order to provide a zinc-phosphate coating with a view to improving the adhesion of the primer-surfacer. Limited experience with sheets which have received this treatment indicate that it introduces some problems with regard to spotwelding that must be solved before the treatment is entirely satisfactory. The zinc phosphate seems to introduce additional resistance in the weld.

The use of all-metal parts in building these bodies permits prefabrication some length of time in advance which is not a feasible method with wood, due to its tendency to warp. The Galvannealed coating on the parts, of course, prevents them from rusting in storage and also in service. When the surface is damaged in service, the Galvannealed coating over the steel has a tendency to prevent the rust from working back under the paint and causing it to flake off, leaving unsightly blotches.

In building these bodies the channel frames and bases are pressed from steel sheets. Also, the tops and supporting bows for the tops are pressed from sheets. Where these bodies are being manufactured by the hundreds, templets are used to insure accurate assembly. In breaking the metal for corners when using Galvannealed material, it is good practice always to leave a radius of at least the thickness of the metal in order to protect the surface. Of course, in using thin metal sheets, it is desirable to do considerable hemming to eliminate sharp edges on the finished parts. This operation can be done satisfactorily although the Galvannealed surface at the edge of the hem may not cover the black metal entirely. In choosing the Galvannealed sheets where the metal will be stretched over dies, it is necessary for easy manufacture to make sure they were made from very soft (deep-drawn) steel sheets. If this precaution is not observed, the Galvannealing process has a

tendency to add to the hardness of the metal and thus interfere with satisfactorily working it in making the shapes.

It is sometimes desirable to attach sheets of rubber to the metal trays or partitions in these all-metal bodies. For this purpose there is now available an excellent type of binder ordinarily designated as "step-tread cement" which is used after cleaning the metal and rubber surfaces with a high-flash naphtha base cleaner. The step-tread cement can be used successfully with black steel sheet, Galvannealed steel, or aluminum.

Now a few words regarding the use of aluminum alloy in body construction. As mentioned previously, this material can be used where the weight saving is of great importance. On the smaller bodies which we are discussing, experience indicates that a weight saving of about 50 per cent can be made. In these bodies, about the same percentage of welding is used as with the steel bodies, and the welds hold satisfactorily. However, again the welding technique must be changed to conform with the requirements of the metal, and it is considerably different from that used for either black sheet steel or Galvannealed steel. A welder accustomed to these metals has a tendency to burn holes in the aluminum metal instead of welding it. In all cases water-cooled points should be used in the welding equipment.

The selection of alloys in the use of aluminum, of course, is important. Present practice indicates that formed sections from 52-S quarter-hard material are satisfactory. For standard sections 17-ST material is being used, and for sheets 3-S half-hard aluminum alloy is satisfactory. On the same design for proper strength the thickness of sheets should be increased from 30 to 40 per cent as compared with steel, depending upon the conditions. The material should be thicker where considerable bending or a possibility of fatigue in the material exists, as compared with somewhat thinner sections where these conditions are not present.

Experience with aluminum bodies indicates that, where this metal is used, there is a greater tendency to mar and wear the surfaces in cases where material is thrown into bins or where tools are hung in such a manner as to strike the side sheets due to car motion while traveling.

The first cost of one of the small-type bodies mentioned previously, when made of aluminum alloy, is roughly 50 per cent higher than for one of Galvannealed steel. The first cost of a Galvannealed steel body, in turn, is approximately 8 per cent higher than one of black sheet.

Although there has been considerable talk with regard to the use of double-strength steels, until very recently they have not been available commercially in thin gages. Where this material has been tried in proper thicknesses for the job, it has shown a marked tendency to spring out of shape when it is attempted to form it, and also the thinner gages used produce very sharp edges which must be cared for. Further experience with this material will doubtless develop satisfactory methods of handling it.

The copper bearing steels seem to have some rust-preventive action. However, with the usual copper content, the rusting is not eliminated to the extent that it is in using Galvannealed sheets.

Light Line-Construction and Maintenance Trucks

As for the bodies used on the vehicles in the mosquito fleet, also on the light-duty line construction and maintenance bodies, all-metal construction is coming rapidly into general use. However, experience indicates that somewhat heavier metal should be used for these side boxes. 18-gage steel seems to be satisfactory. For the side panels and roof 20-gage steel is used. For the generous side boxes on these bodies similar construction and kinds of materials to those outlined previously for the smaller bodies, are employed. The framework for the bodies is made principally of formed sections, and the joints throughout the body very generally are spotwe.ded. Diamond-tread steel stock of approximately 14 gage is suitable for the floor.

The front end of such a body is entirely enclosed as shown in Fig. 9. A generous window is provided in the rear of the cab and no window is placed in the front of the body where an opening registers with the cab window. The space between the cab and body around the window opening is enclosed by cementing and bolting a tarpaulin bellows or by cementing a strip of sponge rubber between the two surfaces.

A folding seat over the winch located in the front end of the body, in addition to providing a place for workmen to ride, encloses this part of the construction. As a further protection for the men riding in the truck, as well as for the tools and material, a telescoping or sliding top may be used so that it can be quickly moved forward out of the way when pole derrick work is to be done or at other times when it may not be needed. In one type the roof slides forward over the cab. In the telescoping roof a piece of tarpaulin material is used between the fixed and sliding-metal sections. In some bodies where a metal roof is not supplied, a strip of tarpaulin is provided for covering the entire top. Up to quite recently 12-oz. colored duck tarpaulin material, wax-treated, was used. The latest practice, however, indicates the superior quality of rubberized fabric for this kind of service. It is more water-proof, has longer life, and shows practically no shrinkage regardless of weather conditions.

On these light construction trucks the general practice now is to cover the sides with sheet metal, rolling it inward at the top in order to provide protection for hanging blocks and tackle as well as to establish a convenient point of attachment for the adjustable metal roof or tarpaulin. Some of these details may not be shown clearly in Fig. 9.

The present tendency is to use wheelhouses on all construction-truck bodies in order to get the platforms as low as practicable for reasons of utility as well as of appearance.

Particularly in the lighter truck units, there is at the present time a considerable variation in CA dimensions in the chassis which are commercially available. This variation, of course, results in difficulty for the designer as well as for the body manufacturer. Also, the body builder should have free space above the chassis frame without interference from helper-spring brackets, ends of transmissions, frame cross-members, or other obstructions.

In addition I should like to suggest something that would be of assistance in the design of bodies, in the line of information which should be supplied on the chassis drawings – for body builders to indicate the dimension from the center of the axle to the top of the frame with metal-to-metal contact between frame member and axle housing. This dimension would assist in planning the wheelhouse heights. The information at the present time is not readily available.

In order to serve the gangs properly, careful study should be given to the compartment and bracket arrangements in the side boxes as well as inside the body for the particular kind of work to be done. As in the smaller-type bodies mentioned previously, with proper body facilities the gang should be able to keep separate the various items at standardized locations on the vehicle to permit quick and easy access to any material or tools which may be required. Even a heavily loaded construction truck should present a very neat and orderly appearance when the tools and materials are allocated to their proper spaces.

For a completely equipped gang truck a water cask is included. These casks are of a thermos type with metal outside and inside cylinders, the inner surface of the inside cylinder

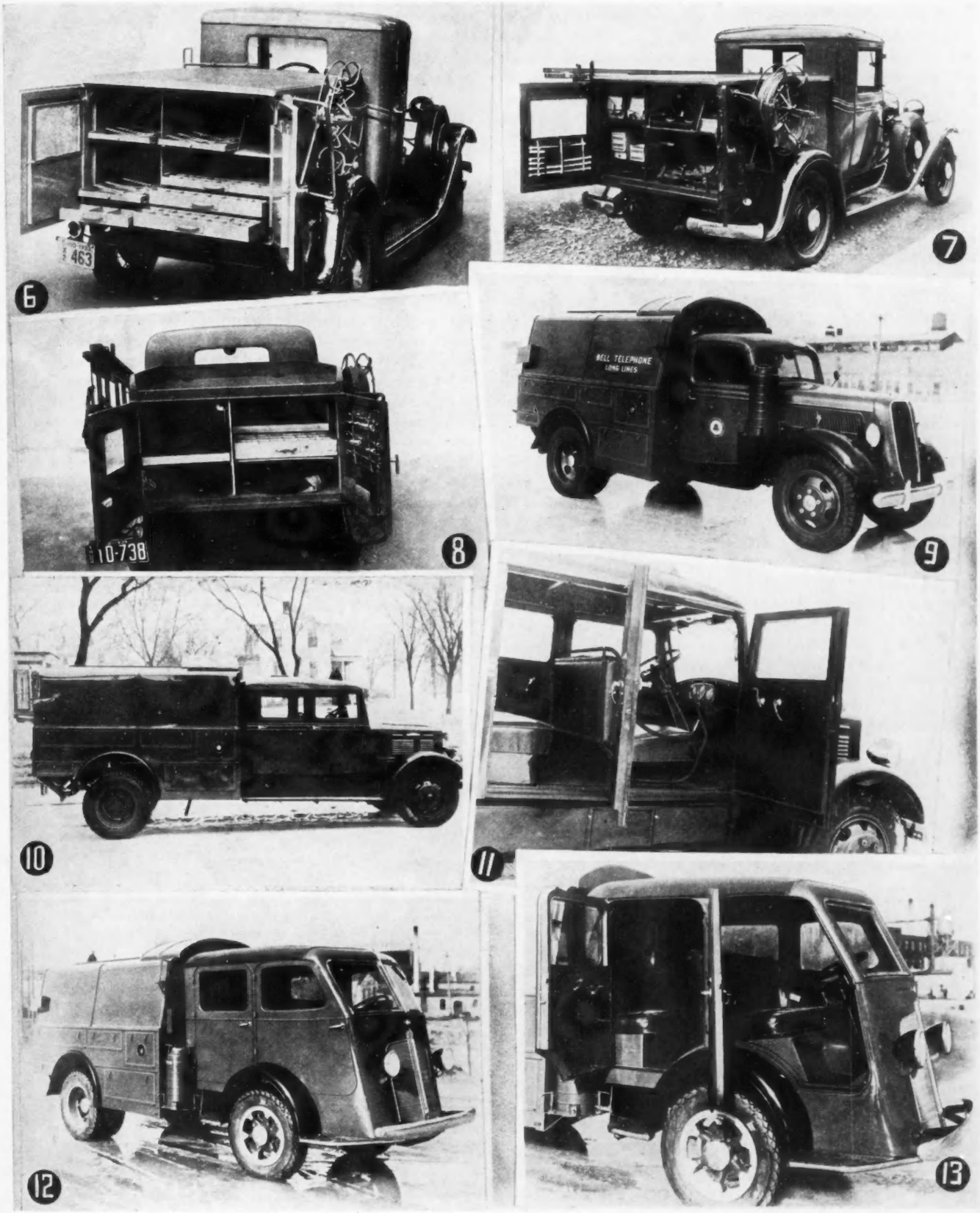


Fig. 6 – Arrangement of Drawers on Light Body Mounted on a "Half-Ton Chassis"

Fig. 7 – Typical Arrangement of Tools in Light Truck Body

Fig. 8 – Light Body Unit Fitted with Special Top for Carrying Bulky Materials

Fig. 9 – Light-Duty Line Construction and Maintenance Truck

Fig. 10 – Light Construction Truck with a Five-Man Cab

Fig. 11 – Steel-Tubing Construction for Front Seat of Five-Man Cab

Fig. 12 – Five-Man Cab on Cab-Forward Unit

Fig. 13 – Interior of Two-Seat Cab-Forward Unit

being lined with porcelain. These casks are available in 5-, 3-, 2-, and 1-gal. sizes, and the size naturally is selected to meet the gang's requirements. On construction trucks there is increasing difficulty in finding a satisfactory location outside the cab on the running board or fender for the water cask so that it is entirely possible that future designs may include space for the casks at a convenient location in the right-front corner of the body with the spigot accessible from the outside.

Heretofore, a spare tire ordinarily has been carried on the left running board of the line-construction trucks. However, there is a growing tendency at the present time to omit the spare tire because of the infrequent tire trouble with present high-grade truck pneumatic tires, and the temporary usefulness of one of the dual rear tires in case of failure of a front tire. The gradually decreasing available space for carrying the spare tire also has some influence upon this matter, it being felt undesirable to carry the tire inside the body where it takes up valuable space.

The light construction truck bodies vary somewhat in size, but there seems to be good reason for not making them any larger than is necessary to carry the tools and equipment which are actually necessary for the day-to-day work of a small gang. Where larger bodies are supplied, the tendency is to overload, which has an unfavorable effect upon the performance as well as maintenance and tends to defeat the economic advantages of using the light inexpensive units. With these points in mind the tendency is to keep dimensions of these bodies near the minimum accepted sizes which are approximately 7 ft. overall width by 9 ft. overall length with approximately 50 in. height from the platform to the top rail of the side panels. A light construction body of this size would have side-box and locker-space capacity above the floor line of approximately 29 cu. ft. In addition to this capacity the under-slung boxes would provide approximately 22 cu. ft. of space. Units of this type in the field ordinarily have gross vehicle weights between 9,500 and 13,500 lb.

Heavy Line Construction and Maintenance Trucks

Heavy construction trucks to serve larger gangs are of the same type as the units just described except that they have greater space capacity. A body for such a unit might be approximately 7½ ft. wide by 10½ ft. long, and the unit might have a gross vehicle weight of from 14,000 to 18,000 lb.

Five- and Seven-Man Cabs – Also C.O.E. Trucks

In the utility field quite recently some use has been made of two-seated, five- or seven-man cabs for gangs which are required to do a considerable amount of traveling in extremely cold weather. With a light construction truck of the type shown in Fig. 10, using a five-man cab, the body is unchanged but the wheelbase is increased by about 30 to 35 in. in order to permit extending the cab back to provide an extra seat for three men in the rear. A cab of 60 in. overall width, which is a commonly used dimension, can be built with free space across the rear seat of about 55 in. which approaches the minimum for three men. The spacing of the rear seat back of the front seat should not be less than 30 in., and toe room should be provided under the rear of the front seat. From the top of the rear cushion to the roof, a space of at least 38 in. should be available.

These cabs on conventional chassis are designed ordinarily with two doors opening at the right side, one opposite each seat, and one door at the left of the driver, with a window at the left end of the rear seat.

The window at the rear of the cab is quite large, and the rear seat can be folded down so that, when the truck driver is operating the winch, he can look through this window and see the top of the winch drum. With this type of cab it also is desirable to have some type of rear-vision-mirror arrangement so that the driver can see back to the top and the upper part of the derrick.

A convenient type of front-seat construction, shown in Fig. 11, is made by using steel-tube framework. This construction provides ample toe room under the front seat for rear-seat passengers and also affords free air circulation which is of assistance in heating the cab in winter.

Similar construction of the five-man cab may be used with the cab-forward units. With this design, illustrated in Fig. 12, practically the same wheelbase is possible as for a one-seated cab on a conventional chassis which arrangement, of course, has the advantage of better maneuverability on the job as well as much easier handling in the garage.

With full appreciation of the design difficulties which have been overcome and of the improvements now under way, it might be helpful to outline from the user's viewpoint some disadvantages of the present designs of cab-forward unit whether of the single-seat or two-seat type. The price is considerably higher than for cabs on conventional chassis. Also, in practically all cab-forward designs there is some sacrifice of space because of the shroud over the engine interfering with the leg room at the middle of the driver's seat, as shown in Fig. 13. In some of the designs this interference is very pronounced. Another difficulty has been found in adapting cab-forward units for winch trucks in the respect that the space underneath the front seat is so congested with motor and transmission, and the space in the cab ordinarily allocated to levers is diminished to such an extent that the winch application is made much more difficult and somewhat more expensive than on the conventional unit. Both single-seat and double-seat cabs of this type are higher and, with the single-seat units, there is no space remaining to carry a spare tire or a water cask, and no room under the seat for the gasoline tank. The bottom line of the projection ahead of the front wheels has a tendency in all designs of this kind to be quite low, which in construction trucks is an objectionable feature in that, when crossing ditches during pole derrick work, for instance, there is the tendency for the front bumper to nose into the ground. I have every confidence that adjustments will be made as more experience is gained to care for these points satisfactorily, and I feel that we are indebted considerably to the manufacturers that are carrying forward this advantageous development.

Referring again to the winch application on the single-seat units only, the transmission projects so far to the rear that there is no space left for a propeller-shaft power take-off which further complicates the winch application.

Underground-Cable Trucks

Where a large amount of underground-cable construction and maintenance work is done in a particular area and certain trucks can be assigned exclusively to this work, it has been found desirable to provide 17,000 to 25,000 lb. gross vehicle weight capacity with much larger bodies than those mentioned previously. Fig. 14 shows such a truck. Where used in congested city streets, cab-forward chassis are convenient. Some bodies designed for this type of work have a width of 7 ft. 8 in., are 13½ ft. in length and approximately 60 in. from the floor to the top rail of the side panels. In general design they are the same as the light- and heavy-duty construction trucks discussed formerly, but in detail they are entirely different due to the different types and quantities of tools and materials which are carried.

The compartments of the side and under-boxes are much larger, as shown in Fig. 15. It will be noted on this picture that the telescoping roof is extended, with the sliding metal member all the way to its rearmost position, and the space between the permanent front metal roof and the sliding part is

covered by rubber tarpaulin material which is fastened permanently to the two roof members. When the rear member is slid forward, it telescopes under the permanent front member.

The total capacity of the side boxes above the floor is about 65 cu. ft., whereas that of the under-boxes is about 49 cu. ft. The CA dimension of 102 in. for a truck of this type permits under-boxes of about 6 ft. in length which, as just noted, materially increases their capacity.

Panel or Van-Type Bodies

Where the delivery of a large amount of rather bulky material is necessary, a van type of body, shown in Fig. 16, may be used, providing ample space and also protection from the weather. It will be noted that although this is a long body, by using the cab-forward type of chassis the wheelbase is not excessive, thus providing the advantages which have been mentioned for cab-over-engine units. The particular body, which is illustrated, is 15½ ft. long, has a 10-ft. CA dimension; yet the wheelbase is only 12 ft.

In order to permit easy loading and unloading under all conditions, full-height side doors may be provided at either side of the body near the front end.

This type of body has an outside width of 7½ ft. and a height above the platform of 85 in. It will be noted that double hinged doors are used at the rear so that these doors, when fully open, swing flush against the sides of the truck as shown in Fig. 17. Also, the hinges are arranged so that the door opening is practically equal to the inside width. Electric lights are supplied in the middle of the truck roof to permit satisfactory operation on dark days or at night where required.

The construction of this body may be of interest since it is made completely of metal-wood. This material gives amply strong construction and saves considerably in the thickness of side walls since the usual inside posts are not required. The sides, front, and doors are made of ⅜-in. material having metal inside and out and three-ply wood in the middle with cotton duck between the wood and metal to allow for expansion. This commercial material of composite form is held together by strong cement. The roof of the body is built of similar material except that the metal covering on the inside is omitted as it is not needed for protection at this location. The metal used in making the metal-wood sheets is Galvannealed steel. Even including all joints the thickness of one side wall can be reduced to less than 1 in. with no ribs or bows to interfere with loading, as compared with 3 or 4 in. for the usual knee-iron and wood-frame construction. Referring to the framework, the corner posts of this body are of angle iron or wood about 2½ in. thick. The cross-bearers are 4-in. channels spaced 14 to 16 in.

As in other types of bodies discussed before, the floor can be made of diamond-tread steel of about 12 gage. This material is fastened to the cross-bearer channels by means of fill-welds which, as is known, are made by drilling holes over the channel flanges through the floor material only, and filling the holes with welding metal thus securely fastening the floor to the bearers.

Body construction of this type has the advantage of about a 15 per cent saving in weight, together with obtaining a maxi-

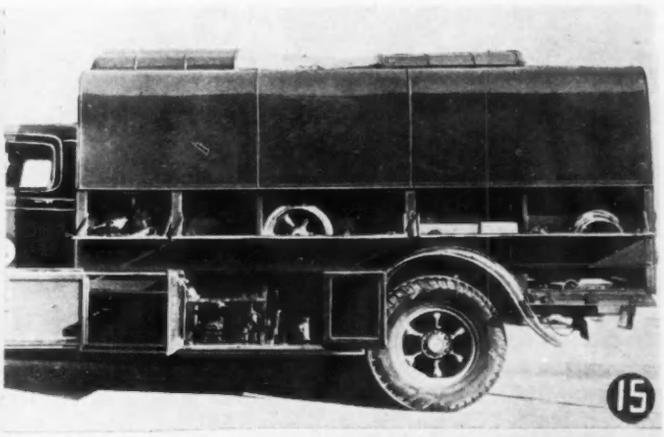


Fig. 14 – Underground-Cable Truck

Fig. 15 – Side View of Underground-Cable Truck

Fig. 16 – Van Type of Body

Fig. 17 – Rear View of Van-Type Unit with Hinged Doors Swung Flush Against the Sides

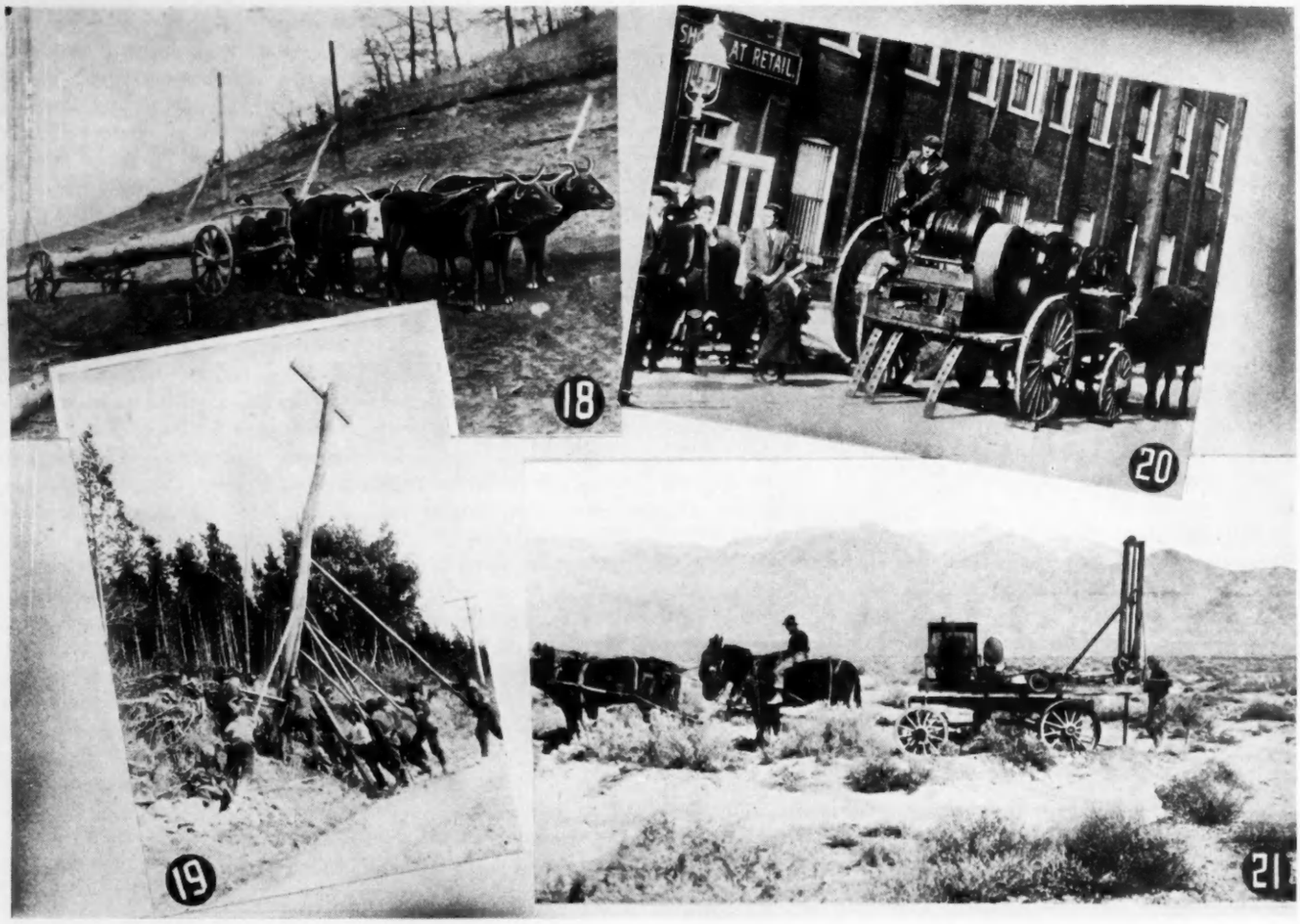


Fig. 18 – Delivering Telephone Poles by Ox Team

Fig. 19 – Early Method of Erecting Poles

Fig. 20 – Early Engine-Driven Winch Pulling-In Underground Cable

Fig. 21 – Primitive Gasoline-Engine-Driven Earth-Boring Machine

mum inside width for a given outside width which may be established by law. Due to the rugged construction and excellent protection of all materials in the body, low maintenance is obtained. As compared with wood, a better paint surface is obtained and the body sides are less wavy than when using plain metal sheets. The metal-covered roof permits repainting it on the same time periods as the metal-covered sides, as compared with possibly twice as frequent repainting when a fabric-covered slat roof is used.

Forerunners of Auxiliary Fleet Equipment

In order to complete the picture I shall now discuss very briefly a few representative auxiliary automotive equipment units applicable to public-utility work. In order to introduce the auxiliary equipment I should like to present a few old treasures which I trust will form a background for the presentation of the up-to-date equipment.

Just 13 years ago I was interested and rather surprised to find poles commonly delivered in one part of the country by means of ox teams as shown in Fig. 18. The method was very slow but I was advised that it was exactly 100 per cent reliable.

The erection of the poles when many of us first entered the public-utility field was a back-testing task for every man in the gang as illustrated in Fig. 19. As many will recall, there was often a slight hesitation at the time the pole was almost up at the vertical balance point when, to the relief of the men on the pikes, it would drop with a thud to the bottom taking with it a considerable amount of earth from the side of the hole.

The desirability of using steel rope of comparatively high strength with engine-driven winches was recognized early as a very desirable mechanism for pulling-in underground cable. The early units consisted of slow-speed, very heavy, single-cylinder horizontal gasoline engines. Fig 20 shows one of these units at work. Although this equipment may seem crude compared with present-day power devices, it constituted a great improvement over the earlier method of pulling by means of manila rope with teams of horses.

It will be noted that, in addition to small chocks to hold the wheels, a type of sprag is used to prevent backward motion of the truck.

Years ago, in connection with building a wire-carrying pole line across the desert, a rather primitive type of gasoline-engine-driven earth-boring machine was used. It is shown in Fig. 21. It consisted of a revolving auger which could be raised out of the hole to remove the dirt as it dug progressive depth sections in making the hole.

Typical Auxiliary Equipment

Now coming to the modern auxiliary automotive equipment, various types of trailers are utilized to good advantage in the utility companies.

Thousands of poles are moved from the yards out to the line or away from locations where they are removed due to decay or obsolescence. The balanced type of trailer shown in Fig. 22 is popular for this work because of its comparatively light weight, ease of handling, particularly in backing, and low cost as compared with a four-wheel trailer.

These trailers may be equipped with an extensible tongue

which is connected to a towing hook built in the rear of the truck when medium and short poles are carried. For transporting very long poles, the first pole loaded on the trailer bolsters is placed in the middle of the load and permitted to project ahead of the rest of the load so that, when the tongue and poles are lashed together properly, this center pole with a suitable drawbar fastened to its front end, serves as the tongue.

A very convenient device has been developed for binding the pole loads with steel rope. It consists of a small hand-operated portable winch which takes up the slack in the binding rope and holds it with a ratchet. Of course, the removable sliding stanchions, two on each side, also help to hold the load in place.

Trailers of this kind weigh from 2900 to 3800 lb. and carry loads of from 3 to 8 tons. Pneumatic tires are now used in keeping with somewhat higher road speeds. Also, electric brakes have been used to a considerable extent. They are operated by a rheostat in the cab convenient to the truck driver. In some instances, vacuum-booster-operated hydraulic brakes have been used. They, too, are operated from the cab by remote control.

Of all the types of power equipment which have been designed and adapted to utility truck work, the most frequently used device is the power winch, and the simplest installation is the single-drum type. Fig. 23 shows a winch of this type mounted on a chassis in the customary manner just before the truck body is installed. The particular winch illustrated has an automatically retained jaw clutch which permits free motion of the drum when disengaged. In this design of winch, if the operator continues to push on the operating lever after he disengages the clutch, a brake is actuated in such a manner that the drum is prevented from spinning too fast as might otherwise occur when the rope is being pulled off. It also will be noted from the picture that his lever is hinged at such a location that the top part may be dropped down below the winch seat for traveling.

By different gear reductions and engine speeds almost any desired rope speed can be secured ranging from 20 to 270 ft. per min. The winch will exert a line pull of 10,000 lb. or even more for a short time. Heavy rope pulls introduce very great pressures between the driving worm and the wheel located, in this case, at the end of the drum. See Fig. 24. In fact, the pressure between these gear teeth may exceed 7 tons at times. In order to withstand these loads satisfactorily, the gears and worms are very accurately cut from high-grade especially selected bronze and steel respectively, and they run in a special extreme-pressure lubricant.

At the end of the worm shaft away from the sprocket there is a safety device in the form of an automatic brake which works when the drum rotates in the reverse direction only, and which will not permit the dropping out of control of a suspended load on the winch line due to overhauling of the gears. This is a very important feature in erecting poles and handling cable reels. It has been found that even though the 29:1 ratio worms are non-overhauling when new, after a period of use has burnished the contacting surfaces, unless a worm brake is used, there is a tendency when lowering a load for it to continue to drop at accelerated speeds after the engine clutch is released. In raising a load with the same worn-in worm and wheel, the load will stop and not overhaul the gears when the clutch is disengaged even though a worm brake is not employed.

In some types of pole derrick operation it is desirable to have a boom line for raising and lowering the derrick and a fall line for lifting the load. For this condition, a double-drum winch is used as shown in Fig. 25. A winch worm brake is shown at the rear end of each of the two worm shafts. Suitable propeller-shaft power takeoffs are available with double-sprocket outlets. A complete double-drum winch, together with its power takeoff, drive chains, mounting brackets and levers, adds about 970 lb. to the weight of a truck. For a single-drum winch, this figure is about 650 lb.

Where a considerable amount of pole work is to be done, the use of earth-boring machinery is advantageous. A truck of the four-wheel-drive type using high-traction tires on all wheels can reach almost any location where it may be desired to dig the hole and set a pole. Fig. 26 shows the machinery mounted on such a truck, with a telescoping derrick. Ordinarily the actual digging of a 6-ft. hole 20 in. in diameter requires from 2 to 3 min. Augers are available in practically any diameter desired.

A telescoping tubular-type derrick is available on one of the types of heavy-duty boring machines as an integral part of the unit. It can be extended or telescoped by power controlled by the earth-boring-machine operator. Also, the usual power-operated leveling worms of the machine are used to tilt or straighten the derrick either sidewise or front and rear. Fig. 27 shows the derrick at work. These power adjustments of the machine and derrick provide features which are very convenient when erecting poles in wire lines or among tree limbs.

The earth-boring-machine units are kept as light as practicable in order to improve their maneuverability across ditches. It will be noted that the body has only skeleton construction for the side panels as shown in Figs. 26 and 27. In some types of operation the unit is used exclusively for pole-setting work, in which case practically no extra tools and materials need to be carried.

When it is desired to travel a short distance between pole holes, ordinarily the derrick is telescoped and carried in the erected position.

However, when moves of any considerable distance are contemplated, the derrick is telescoped by power and then one of the power-operated leveling worms lowers the entire top of the machine into the truck body as shown in Fig. 28.

Recently much interest has been shown in, and a considerable amount of experimental field work done with, a rather novel type of ladder-platform. This equipment consists of a specially designed, very strongly built extension ladder having a platform near its top where one or two men can work. The ladder is always carried and supported in the truck. When in use, its bottom is pinned to the floor in a swivel joint so that, before the man ascends, he can readily swing the ladder around to any position on the sides or back of the truck, where it is supported by a strong steel ring permanently mounted on the truck body. The ladder also is arranged so that it can be extended or shortened by one man.

This device ordinarily is used for inspection and repair of aerial plants, tree trimming, and so on, as shown in Fig. 29. In order to coordinate the work properly a telephone connection may be installed between the man on the platform and the driver in the truck cab. In addition to this safety precaution, it is also very desirable that some mirror arrangement be provided so that the truck driver can watch the ladder and platform as he moves the truck along very slowly.

To take down the ladder for moving, it is first telescoped down to its shortest length and swung around to rest on a straight bar on the front side of and below the supporting ring. Then the pivot pin is pulled from the floor and the ladder balanced on the cross-bar after which it is lowered to a horizontal position and clamped in place as shown in Fig. 30. It will be seen that, in this position, the ladder-platform unit does not interfere with body space and, if desired, the telescoping roof can be pulled back beneath the ladder.

The permanent ring is attached to the framework of the body and adds about 200 lb. of weight. The ladder platform weighs approximately 214 lb.

Fig. 22 – Balanced Type of Trailer for Carrying Telephone Poles

Fig. 23 – Single-Drum Power Winch Mounted on a Truck Chassis Before the Body Is Installed

Fig. 24 – Worm and Worm Wheel of Power Winch

Fig. 25 – Double-Drum Power Winch Mounted on Truck Chassis

Fig. 26 – Earth-Boring Machinery and Telescoping Derrick Mounted on Four-Wheel-Drive Truck

Fig. 27 – Telescoping Tubular-Type Derrick at Work

Fig. 28 – Earth-Boring Machine and Derrick Lowered into Position in Truck Body for Long Moves

Fig. 29 – Extension Ladder Platform Applied to Tree Inspection

Fig. 30 – Method of Clamping Ladder-Platform Unit in Place When Not in Use

Conclusion

Now that we have reviewed features of some of the representative units of a utility fleet, it might be of interest to note a few general objectives an engineer or executive might wish to consider in rebuilding his fleet, as this rebuilding occurs progressively in the operation of retiring old vehicles and adding new ones.

Each type of vehicle should, it seems, incorporate features which adapt it in the best possible manner to serve the gang for which it is intended. This requirement is evident when it is realized that supplying the best available equipment costs only perhaps 10 to 20 per cent of the daily cost of the gang. Any apparent saving in curtailing the desirable features of the automotive equipment may be lost many times over through lowered gang efficiency.

Generally speaking, suitable types of equipment which contribute most to efficiency also add to the safety of the operations as well as the convenience of the workmen. These very important items should be studied carefully in connection with every feature considered. Also it should be kept in mind that much can be done in planning the design of various details of the truck equipment and toward facilitating the manufacture with a corresponding reflection upon price and delivery.

The appearance of each vehicle on the highways bearing a company's name may or may not have importance from an advertising viewpoint. I may not be qualified to comment upon this matter, but I do know that a fleet of cars so designed and maintained as to present a pleasing appearance does elicit many favorable comments, and vice versa. Also we should not lose sight of the favorable psychological effect upon the workmen, of having a truck which presents a pleasing appearance with an orderly arrangement of all the tools and materials.

To the three factors just mentioned I should like to add a fourth desirable consideration. This factor probably lies half way between safety and advertising. It has to do with planning the vehicles so that in size, ability, and operating features they will not interfere unnecessarily with or be objectionable to other users of the highways.

Cooperation of the various users among themselves and with the manufacturers of the equipment, showing mutually sympathetic regard for each other's problems, will accomplish much in approaching these ideals.

C. O. E. Vs. Conventional Truck Maintenance

I THINK there does exist a general impression, and this impression is understandable until a thorough analysis is made, that the cab-over-engine vehicle must cost a lot more to maintain. It is very difficult to visualize how one can take an engine and practically hide it, as we have to in this type of vehicle, and still preserve a measure of accessibility that normally seems to exist in conventional trucks. I say "seems to exist" because in some conventional trucks extremely stylized designs in many cases have made the accessibility of these conventional trucks a matter of much argument. However, if there is a higher maintenance cost on the C.O.E. type, it is most certainly one of small degree.

I think also we should not entirely lose sight of the fact that, knowing this type of vehicle would meet with some skepticism with operators as to its accessibility and maintenance cost, more than special care has been paid to design details to meet these objections. Whilst this construction has involved increased cost originally, the ultimate satisfaction and reduction in maintenance costs seem to justify the special design features.

Generalizing, I think we may say that, in the average C.O.E. model, it is possible to expose the engine for ordinary adjustments of distributor, changing of spark-plugs, attention to water pump, and the normal accessories of the engine in about the same number of motions as on a conventional job. Actually, in the case of units of a rating where the tire size demands high fenders, the accessibility of the conventional engine for these engine details is very often not as good as the accessibility inside the cab of the C.O.E. Further, I think it may be argued reasonably on the matter of removal of powerplant, which in the conventional job involves moving of hood and radiator before the engine can be swung out, that it is certainly not much easier than where an engine is mounted on a sub-frame and slid out from the front with the radiator and transmission as part of the general engine assembly.

An operator of rather large experience brought out the point that the design feature of being able to slide the engine out had very worthwhile advantages. He felt that, after any maintenance department was familiar with the details incident to this operation, it could be performed very efficiently and the disconnecting of controls was easier in some cases.

With regard to the sheet-metal work and the cab, certainly in present-day designs there should be no reason to expect as high maintenance on the C.O.E. job because of the compactness of the sheet metal and cab set-up. Fender maintenance, from the point of view of cracking or breaking under severe operating conditions, is entirely unknown in the C.O.E. truck whereas in the conventional job it is often the case that, under repeated severe distortion, cracking of fenders occurs.

If we take into account the maintenance expense resulting from accidents, the C.O.E., due to its improved vision from a driver's standpoint, should certainly show up in a favorable light.

Based on the study of design features of C.O.E.'s and on the considered opinions of many operators, it would seem that whatever compromises have been necessary in the C.O.E. design have not been of such a nature as to increase seriously the maintenance cost. It is obvious that one year from now we shall have a much clearer picture of the differences between the two vehicles from the maintenance angle. At the present time it would seem reasonable that the general utility, short overall length, and other C.O.E. features should be weighed somewhat against the cost of maintenance. Furthermore, when it passes through the same range of design development as has the conventional truck, we can expect a further improvement in the maintenance costs of this type of vehicle.

Excerpts from the paper of the same title by R. Cass, presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 5, 1937.

Cast Camshafts and Crankshafts

By F. J. Walls
The International Nickel Co., Inc.

OUR old conception of cast iron as being an unreliable, weak, brittle metal has been changed by the coordinate efforts of engineers, metallurgists, and founders. We now have a versatile material possessing virtues in designs where dependability and durability are primary requisites. The crankshaft, which converts the reciprocal motion from the power-producing factors of an internal-combustion engine into rotary action and the camshaft, which controls these factors, must be reliable and economical to an extent equal to the wrought materials which they have replaced if they are to be adopted in modern production.

It is evident from the quantity now in use and the present daily consumption that both of these important parts made from cast material are performing satisfactorily.

It would require a lengthy volume if one were to cover in detail the development of cast cams and cast cranks, but any statement should not be acceptable unless it pays tribute to those foundrymen, metallurgists, and engineers whose foresight, courage, and continued experimentation have made the present volume production a pleasant reality. The Campbell, Wyant & Cannon Foundry Co., the Caterpillar Tractor Co., the Ford Motor Co., and many others justly deserve words of praise for their contributions in elevating cast ferrous materials to the status which they now enjoy.

The natural question which production engineers ask is: "What are the factors that justify the substitution of cast materials for forged materials in camshafts and crankshafts?" These factors may be classified under the following general headings: economy, engineering design, and material characteristics.

From the standpoint of economy the initial outlay is reduced considerably in the casting process as compared to the forging process. Pattern costs, as a rule, will be several times less than the cost of a die and, in the case of new designs where changes are sometimes found necessary, pattern changes are made readily whereas changes in dies are usually difficult, if not impossible, to make. Expensive forging equipment and heat-treating furnaces are not needed for cast materials. Reduction in machining costs made possible by the use of cast materials has been described in the literature quite extensively and need not be touched upon in this discussion.

From the standpoint of engineering design it may be said that the engineer has been presented with possibilities which he has often dreamed of, but which could not be carried out in forging designs. By the casting process it is possible for him to locate his counterweights and control gravity centers at those points where his mathematical calculations show that they should be placed. Crankpins, bearings, and cheeks may be cored out, thus eliminating a great deal of unnecessary weight, which is often very expensive to remove from a forging. By proper design this elimination of weight oftentimes increases the engine output by the reduction of physical forces.

The improvements in characteristics of cast materials through the use of alloys, metallurgical control of melting operations, improved molding conditions, and technical control in the foundry have taken the guesswork out of the casting process and made the uniformity of production more certain and dependable. Better knowledge of the physical properties of this improved material is probably the most important factor pertaining to the substitution of cast materials for forgings. This knowledge has been brought about through organized research and unlimited practical tests.

The physical properties which are so essential in cast cams and cast cranks may be summed up as follows: rigidity, wear resistance, torsional strength, fatigue strength, and damping capacity. I wish to touch briefly on a few of these properties which are being explored thoroughly at the present time.

Fatigue Properties

It is evident, if one reviews the literature, that the fatigue properties of high-strength cast irons have not been explored sufficiently in order to set up as an engineering standard an endurance limit having a definite ratio with respect to tensile strength. This factor has been established tentatively by various investigators as ranging from 40 to 55 per cent of the tensile. Recent investigations, however, seem to indicate that the endurance limit of the 50,000 and 60,000 lb. per sq. in. class irons can be assumed to be 50 per cent of the ultimate tensile strength. There are further indications that the structure and chemical composition have an important bearing on the endurance limit.

If it is true that the endurance limit varies linearly with the tensile strength, then we may conclude that an iron having a tensile strength of 50,000 lb. per sq. in., has an endurance limit of about 50 per cent, or 25,000 lb. per sq. in.; therefore, an iron having a tensile strength in the neighborhood of 70,000 lb. per sq. in., would have an endurance limit of approximately 35,000 lb. per sq. in.

The sensitivity to notch effect on cast iron has been reported to show a reduction in the endurance limit of from 5 to 25 per cent, depending somewhat on the shape of the notch. Two notch fatigue tests using a square notch, as I will outline later, show a reduction in the endurance limit of approximately 10 per cent, which is considerably lower than that exhibited by steel. It is this low sensitivity to notch effect that places cast iron in a position to compete with a material whose higher initial strength and endurance limit, in a smooth specimen, is reduced to the same level, or even lower when square corners, notches, grooves, and stress-concentrating centers such as oil holes are induced.

Fatigue tests on two different nickel-molybdenum gray irons made in the cupola are shown in Figs. 1 and 2, together with

[This paper was presented at the New England Section Meeting of the Society, Cambridge, Mass., Feb. 9, 1937.]

the composition and other physical properties of these irons. The gray-iron test specimens were cut from arbitration bars in a section half-way between the center and the edge, and are of a modified Moore type having overall a length of 3 7/16 in., a maximum diameter of 0.480 in. at the shoulders, and a reduced section of 0.300 in. diameter. The test section is cut to a radius of 3 53/64 in. The notched specimens have a square notch which has a width of 3/16 in. and a reduced section of 0.300 in. (maximum section is 0.480 in.). The squareness of each notch is observed on the microscope by projecting its shadow on the ground glass before testing. The tests were conducted in R. R. Moore rotating-beam machines revolving at 1750 and 3500 r.p.m. Properties of crankshaft irons from two different heats are given in Table 1.

Table 1 – Properties of Crankshaft Irons

Type	Ni. Mo.	Ni. Mo.
Heat No.	3386	3387
Furnace	Cupola	Cupola
Total Carbon, per cent	3.04	3.06
Silicon, per cent	2.07	2.17
Manganese, per cent	0.45	0.62
Nickel, per cent	2.03	1.70
Molybdenum, per cent	0.34	0.62
Brinell Hardness No.	277	241
Tensile Strength, lb. per sq. in.	52,450	52,900
Modulus, lb. per sq. in.	18.37×10^6	18.45×10^6
Fatigue Limits		
Notched, lb. per sq. in.	23,500	25,500*
Smooth, lb. per sq. in.	27,000	27,500*
Fatigue Ratios		
Notched	0.45	0.49
Smooth	0.515	0.52
Transverse Strength, lb.	5905	6205
Transverse Deflection, in.	0.157	0.212

* Approximately.

Vibration Damping

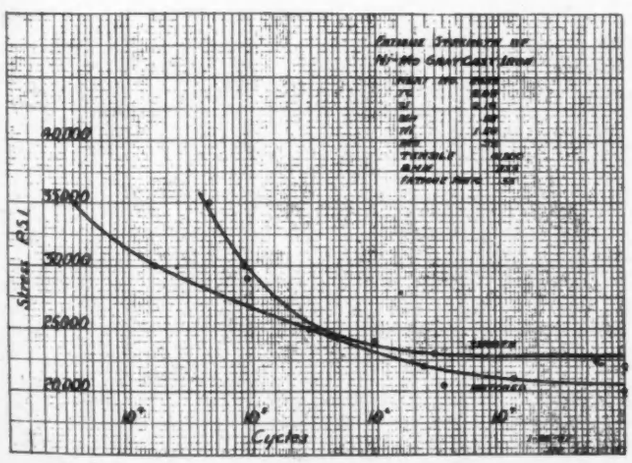
"Damping capacity" is defined as the amount of work dissipated into heat by a unit volume of the material during a completely reversed cycle of unit stress. It is the energy lost in the conversion of potential into kinetic energy and measured in inch-pounds per cubic inch per cycle. The "specific" damping capacity which is used in comparing materials of different moduli of elasticity is calculated from data obtained from testing a solid round bar machined from the material in question to the following dimensions in a Foepple-Pertz damping-testing machine: 11¾ in. long, reduced section 6½ x ½ in. in diameter, with square ends.

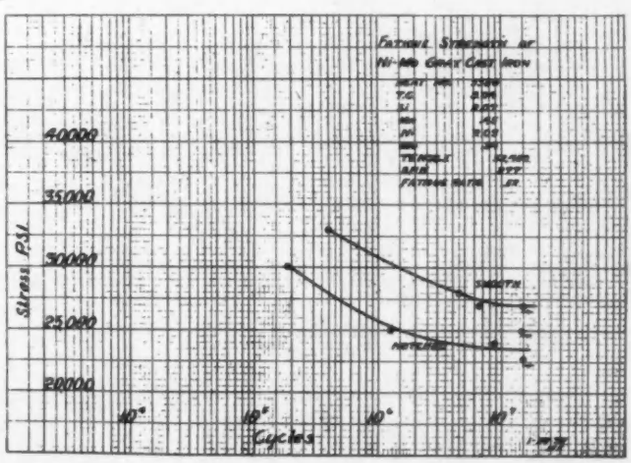
This free-vibration method of determining specific damping capacity and other methods are ably described in a paper: "Damping Capacity of Materials," by G. S. Von Heydekampt, published in Vol. 31, 1931, of the *Proceedings of the American Society for Testing Materials,* pp. 157-170. In his discussion he states that "The damping method seems to furnish a novel and promising tool for investigating the range of low stresses which is the one mostly used in service." He applies the name "dynamic ductility" for the damping capacity to distinguish it from static ductility or elongation since both are a measure of plastic behavior.

He further states: "The dynamic ductility as measured by the damping capacity seems to be related to the 'tenderness' effect. In other words, a material of higher damping capacity is less sensitive to the influence of surface notches or sudden changes in cross-section, which strongly affect the fatigue strength of 'dynamically' brittle materials. Materials having high damping values are able to be deformed to a higher degree than Hooke's law predicts without being damaged. In this way 'stress raisers' may be compensated for by internal yielding, the amount of which is measured by damping capacity."

In a paper by G. R. Brophy and E. R. Parker published in *Transactions of the American Society for Metals,* December, 1936, pp. 919-931, entitled "Damping Capacity – Its Variation and Relation to Other Physical Properties," the authors state that "In any metal damping capacity may vary greatly depending upon such factors as stress, chemical composition, structure, heat-treatment, mechanical work, and especially temperature. Combinations of these factors, which are usually encountered, complicate the interpretation of damping-capacity determinations until the influence of each is understood." Their conclusion is as follows: "The study of damping capacity as a property of metals is, to say the least, extremely interesting and offers a means of understanding better the action of metals under stress. It is not proposed here that the damping test be used as a substitute for others, but in conjunction with them, and possibly as a means of surveying rapidly a large number of materials and thereby eliminating long and tedious tests on the unpromising.

"As a property, it is of greatest importance in the control of the deflection and whipping of rotating shafts, and the surging of springs. Its variation with temperature appears to be a controlling factor in the fatigue and creep resistance of steel. Other relations undoubtedly will be found as there remain a number of problems to be investigated; for instance, a study





Figs. 1 and 2 – Fatigue Strength of Two Nickel-Molybdenum Gray Cast Irons Made in the Cupola

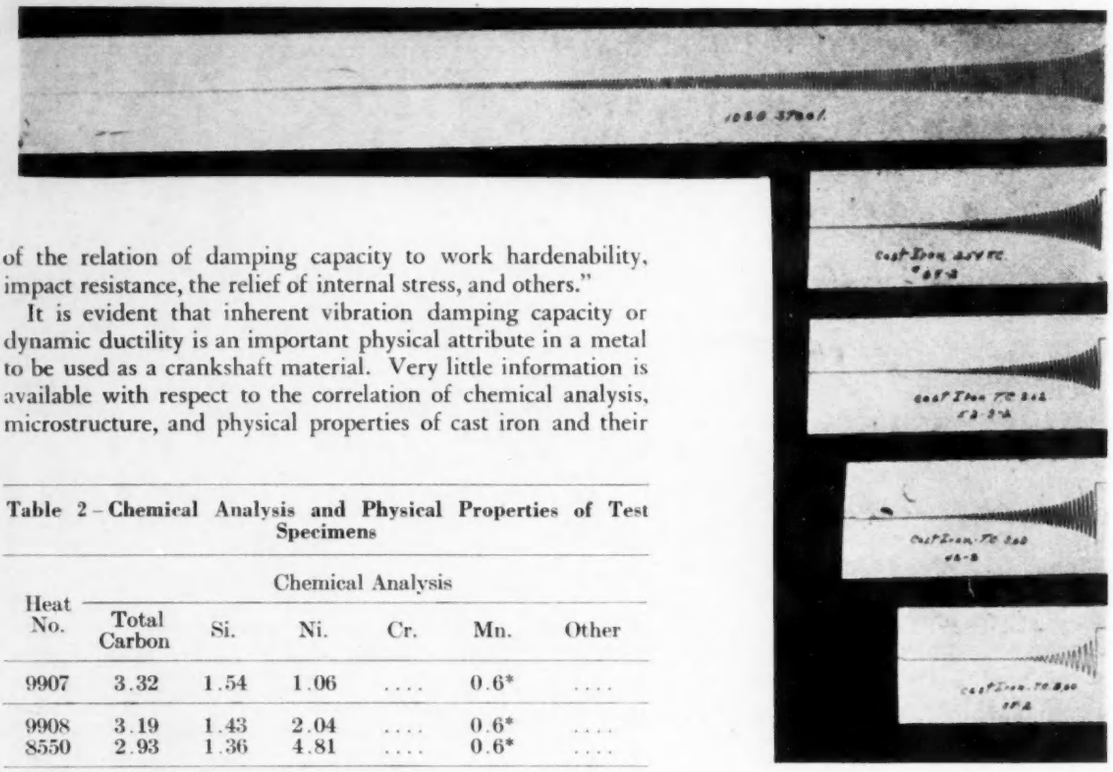


Fig. 3 – Comparative Damping Curves of S.A.E. 1020 Steel and Four Cast Irons

of the relation of damping capacity to work hardenability, impact resistance, the relief of internal stress, and others."

It is evident that inherent vibration damping capacity or dynamic ductility is an important physical attribute in a metal to be used as a crankshaft material. Very little information is available with respect to the correlation of chemical analysis, microstructure, and physical properties of cast iron and their influences on the inherent damping characteristics. A comparison of vibration damping curves taken from steel and cast irons, shown in Fig. 3, is interesting from a practical standpoint, if not academically, in so far as it shows the relative merits of the various materials. It is hoped that, in the near future, work that is now in progress will be made available in the literature.

Table 2 – Chemical Analysis and Physical Properties of Test Specimens

Heat No.	Chemical Analysis					
	Total Carbon	Si.	Ni.	Cr.	Mn.	Other
9907	3.32	1.54	1.06	....	0.6*	....
9908	3.19	1.43	2.04	....	0.6*	....
8550	2.93	1.36	4.81	....	0.6*	....
8568	3.22	1.72	1.20	0.51	0.6*	....
9910	3.24	1.41	1.30	0.21	0.43	....
3342	3.20	1.20	1.50	....	0.6*	0.42 Mo.
3372	2.76	2.07	1.57	....	0.75	0.69 Mo.
8613	2.39	1.77	1.80	....	0.7*	0.72 Mo.
8606	2.56	1.37	1.97	....	0.72	0.75 Mo.

* Approximately.

Table 2 (Continued)

Heat No.	Arbitration Bar Transverse Test, 12-in. Span		Arbitration Bar Brinell Hardness No.	Ultimate Tensile Strength, lb. per sq. in.	Torsion Tests			Shearing Modulus of Elasticity, lb. per sq. in. x 10^6	Permanent Angle of Twist, deg. per in.	Ratio of Torsional Proportional Limit to Ultimate Strength
	Stress, lb.	Deflection, in.			Proportional Limit, lb. per sq. in.	Yield Stress, lb. per sq. in.	Ultimate Strength, lb. per sq. in.			
9907	4525	0.17	212	38,600	16,980	30,580	50,250 52,300	5.40	3.82 4.20	0.338
9908	3700	0.13	212	39,280	18,200	33,380	54,450 53,000	. .50	3.68 0.70	0.334
8550	3515	0.11	331	41,400	20,000	37,000	57,900 58,000	5.33	2.51 2.20	0.346
8568	4720	0.15	248	40,000	17,600 13,950	34,700 34,350	54,200 54,600 55,300	5.15 5.825	5.36 2.68 2.80	0.324 0.255
9910	4540	0.16	212	36,860	20,600	31,290	49,500 47,000	5.138	3.83 2.10	0.416
3342	5500	0.19	241	43,420	19,400	37,000	58,800 54,200	6.215	1.92 2.10	0.330
3372	6090	0.18	255	54,100	15,780 16,980	40,000 44,000	74,000 84,500	7.16 7.00	5.61 4.60	0.213 0.201
8613	5210	0.12	293	50,500	47,900 47,400	63,500 72,000	63,500 72,000	8.84 8.13	1.92 2.05	0.754 0.659
8606	5380	0.15	293	51,390	33,980 40,100	62,700 64,800	75,700 70,600	7.56 7.11	1.15 1.15	0.449 0.568

For the present we can only assume that a crankshaft made of a material of high damping capacity with moderate strength may have a greater effective strength and life expectancy than one made from a material with much higher strength having a low damping capacity. Designers have eliminated the critical torsional oscillations to a certain extent by means of mechanical vibration dampers properly installed in the engine, but these devices usually are only efficient at normal operating speeds and, where an engine is operated at a higher critical period for a sufficient length of time, resonance failure is more liable to take place in a material of low damping capacity than one of high damping capacity. Where the impulse energy is restricted to small amplitudes by internal-frictional dissipation well within the fatigue range, failure will not be likely to occur.

Torsional Strength

Much of the current development of cast iron for applications such as crankshafts and camshafts has been carried on with little specific information on the torsional properties of this material. Figs. 4, 5, and 6 show graphically the torsional properties in comparison with the tensile strength of some 20 different irons which were investigated recently. The composition and physical properties as obtained on 9 of these irons are listed in Table 2.

Composition apparently affects the torsional strength in the same way that it affects tensile strength. A straight-line rela-

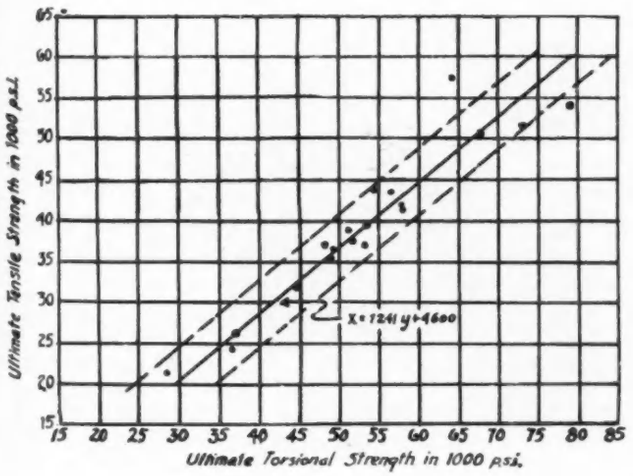


Fig. 4 – Relation Between Ultimate Tensile Strength and Ultimate Torsional Strength

tion has been established which may be obtained by the equation:

$$x = 1.241\ y + 4600$$

where x = ultimate torsional strength.
and y = ultimate tensile strength.

As shown in Table 2, a high shearing modulus of elasticity is associated with high torsional strength.

The torsion-test specimens were machined from arbitration bars with the reduced section approximately 5 in. long and 0.750 in. in diameter. Standard methods of testing and standard formulas for calculating the various physical properties were used. The relatively high values of cast iron in torsion are interesting from the standpoint of crankshaft design.

Camshafts

There are several methods from a foundry standpoint that might be used in making cams, but the two most successful methods are differential hardening and controlled chill, both of

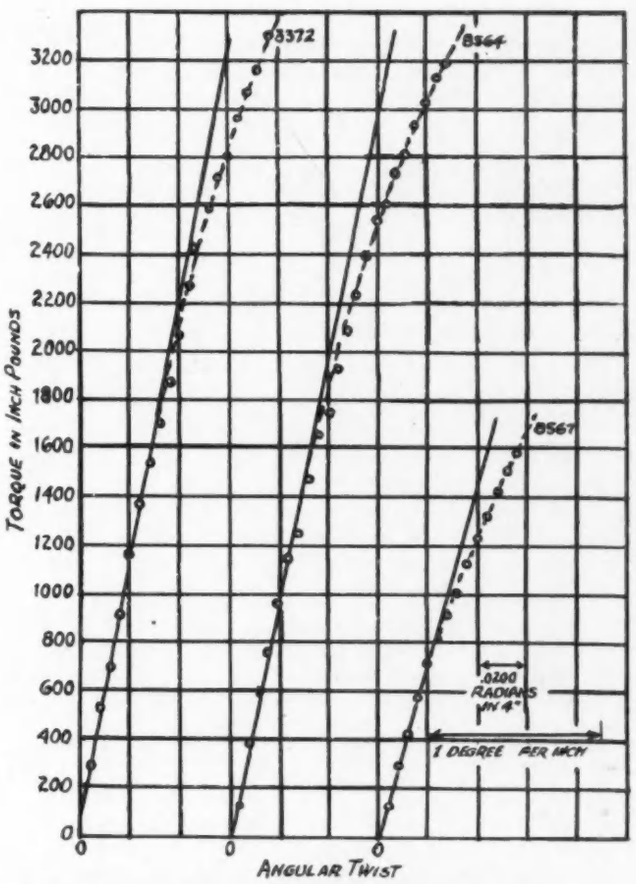


Fig. 5 – Typical Stress-Strain Curves Obtained in Torsion Tests on Cast Irons

which have been written up in the literature quite extensively, but may be outlined briefly as follows:

The first method was developed by the Campbell, Wyant &

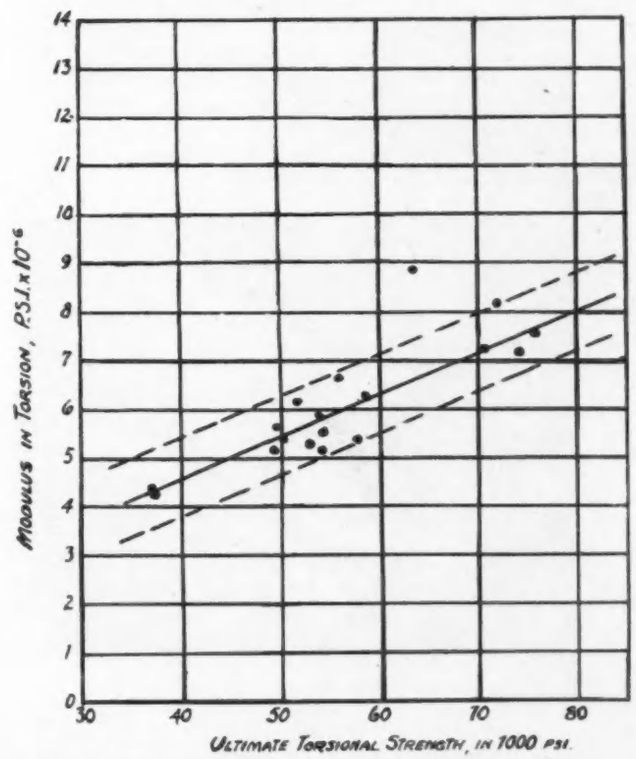


Fig. 6 – Relation Between Torsional Modulus and Ultimate Torsional Strength

Cannon Foundry Co., whose present daily production amounts to approximately 15,000 automotive cams of several different designs. After several years' experimenting they selected a composition as follows:

Total Carbon	3.10-3.40
Silicon	2.10-2.40
Sulphur	0.10-maximum
Phosphorus	0.20-maximum
Manganese	0.50-0.75
Chromium	0.75-1.00
Nickel	0.20-0.40
Molybdenum	0.40-0.60

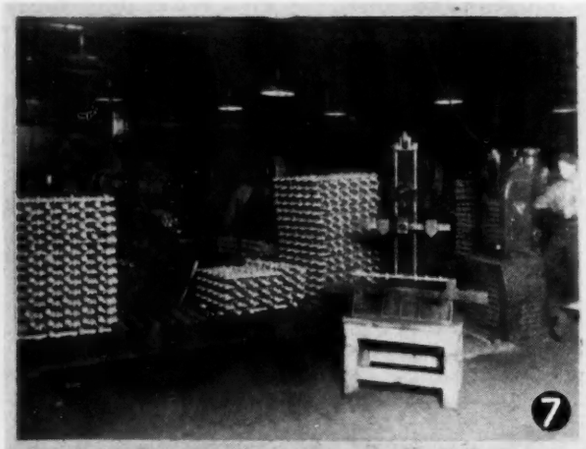
This metal, called "Proferall" (processed ferrous-alloyed iron) is a duplexed cupola-electric furnace iron having a Brinell hardness as cast in green sand of 262 to 293. The camshafts are given an impact test by placing them in a fixture supporting the shaft from each end, and dropping a 35-lb. weight a distance of 14 in., striking the shaft in two places as shown in Fig. 7. Hardness on the gear blanks and bearings are checked on a Gogan and file tests made on the cams after which they are centered and straightened. Finishes and hardness are shown in Fig. 8.

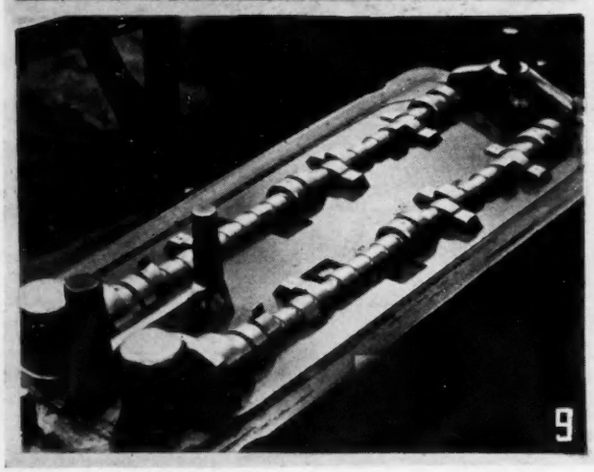
The hardening operation on the nose of the cams may be carried out by either the flame method or the high-frequency method, details and patents of which are owned by Campbell, Wyant & Cannon. The principle involved is to heat the part above the critical rapidly, and allow sufficient time for penetration of the heat to the desired depth and, by rapid cooling, a martensitic structure is obtained on the nose as shown in Fig. 8 having a Scleroscope hardness of 65 minimum.

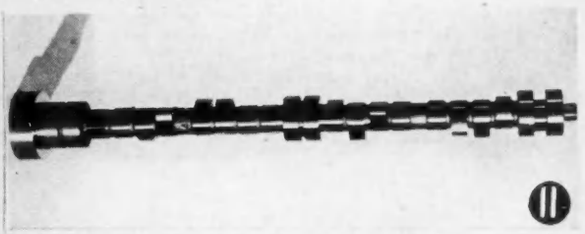
The second or "controlled-chill" method was developed by the Ford Motor Co., using an iron melted in the cupola having the following composition:

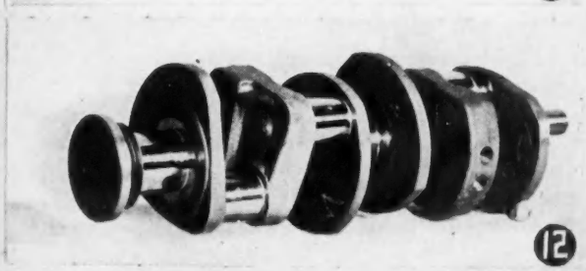
Total Carbon	3.30-3.65
Silicon	0.45-0.55
Manganese	0.15-0.35
Copper	2.50-3.00
Chromium	0.00-0.25

These cams are produced two in a mold in green sand without chills and gated as shown in Fig. 9. The casting is mostly gray with the cam tips white for a specified depth as illustrated in Fig. 10. Accurate metallurgical control of the melting and pouring operations as developed by Ford metallurgists has made this method very successful and economical. Adjustments in the composition of the molten metal are made at the cupola in 1000-lb. ladles when found necessary. The necessity is determined by pouring 1¼-in. round test bars 6-in. long in dry sand molds, breaking them and examining the fracture. Ferro-silicon or ferro-chromium is added to decrease or increase, respectively, the chill depth on the nose of the cams which must be held to within ⅛-in. It is claimed that the high percentage of copper lessens the usual tendency of variable chilling capacities, and that a minimum amount of ladle "doctoring" is required due to its stabilizing effect. Rigid shafts









Figs. 9 to 12 Courtesy Ford Motor Co.
Fig. 7 Courtesy Campbell, Wyant and Cannon Foundry Co.

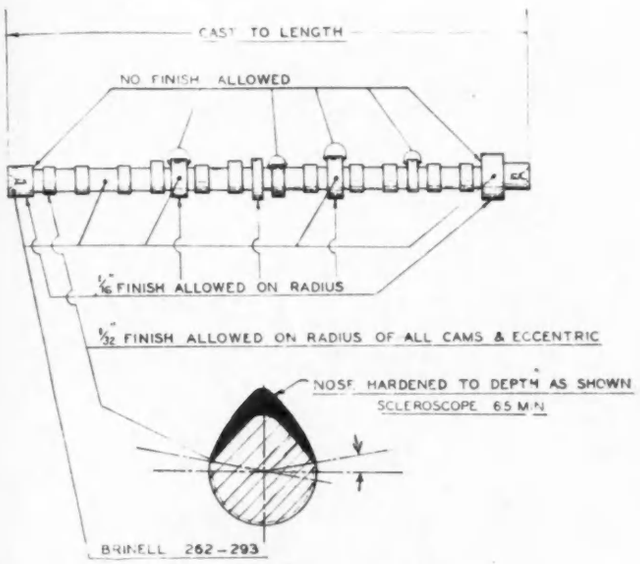
Fig. 7 – Impact Testing of Cast Camshafts

Fig. 9 – Pattern Showing Gating of Ford Camshafts

Fig. 10 – Fracture of Ford Camshaft Shown in White and Gray Sections

Fig. 11 – Ford Camshaft

Fig. 12 – Ford Crankshaft


CAST TO LENGTH
NO FINISH ALLOWED
1/16" FINISH ALLOWED ON RADIUS
1/32" FINISH ALLOWED ON RADIUS OF ALL CAMS & ECCENTRIC
NOSE HARDENED TO DEPTH AS SHOWN
SCLEROSCOPE 65 MIN
BRINELL 262-293

Courtesy Campbell, Wyant & Cannon Foundry Co.

Fig. 8 – Diagrammatic Representation of Finish on Various Parts of Camshaft and Hardness in Section

without a measurable amount of wear are produced by this method; one of them is shown in Fig. 11.

In addition to the two methods outlined previously, there are others that should be mentioned. The use of chills in the mold, either rammed in or inserted in prints, to produce white iron on the cam lift and nose and a composition suitable to obtain gray sections in the parts requiring machining operations, was the first method used in casting cams and is still used where the quantities are small.

Another method is to select an alloyed iron that will cast gray with good machinability and that will respond to quench-hardening. This method is used in designs where susceptibility to warpage is not likely to be encountered. Air-hardening alloyed irons have been developed, and castings made of these irons are less likely to warp due to lower differential temperatures.

Crankshafts

Crankshafts have been made from all kinds of materials, both in the forged and in the cast form, but not until recently were the cast products considered safe for use in high-speed engines. The Ford Motor Co., after lengthy and costly experiments in the laboratory and in service, adopted an alloy cast-steel crankshaft shown in Fig. 12. Between two and three million of these shafts have gone into service with a marked reduction in failures below those encountered with forgings. The merits of the cast-steel shaft over the forged one are said to be: (1) economy of manufacture due to eight less fabricating operations and the removal of 9 lb. of metal as compared to 24 lb. (2) improvement in engine performance due to a decrease in weight from 66 lb. to 56 lb. in the finished shaft and a lower coefficient of friction (3) twice the resistance to fatigue failures, probably due to a lower notch sensitivity.

The composition and physical properties of Ford cast-alloy steel, which is melted in the electric furnace, are as follows:

Total Carbon	1.35-1.60
Manganese	0.60-0.80
Silicon	0.85-1.10
Chromium	0.40-0.50
Copper	1.50-2.00

Heat-treatment:

Heat to 1650 deg. fahr. hold for 20 min. Air quench to a minimum of 1200 deg. fahr. Reheat to 1480 deg. fahr. and hold 1 hr. Cool in furnace to 1000 deg. fahr. in 1 hr.

Physical Properties after Heat-Treatment:

Tensile strength, 95,000 to 110,000 lb. per sq. in. Elongation, 2.5 to 3 per cent. Brinell hardness, 250-320.

At the present time nickel-molybdenum gray cast irons have been adopted as standard by several manufacturers. The significance of this development can be emphasized best by showing illustrations of a number of cast crankshafts from which the diversity in design and size may be observed. They are shown in Figs. 13-18 inclusive.

Some of these crankshafts have been in service for three years or more, whereas others are running in experimental engines with very satisfactory results. Campbell, Wyant & Cannon Foundry Co., began this development prior to 1930 but, since this time, it has progressed rapidly. To date, several thousand crankshafts produced by them have gone into nearly every type of service including automobiles, trucks, tractors, marine, and stationary engines.

This metal known as Proferall 3X (processed ferrous alloyed) is made either from metal duplexed in the electric furnace from the cupola or cold-melted in an electric furnace:

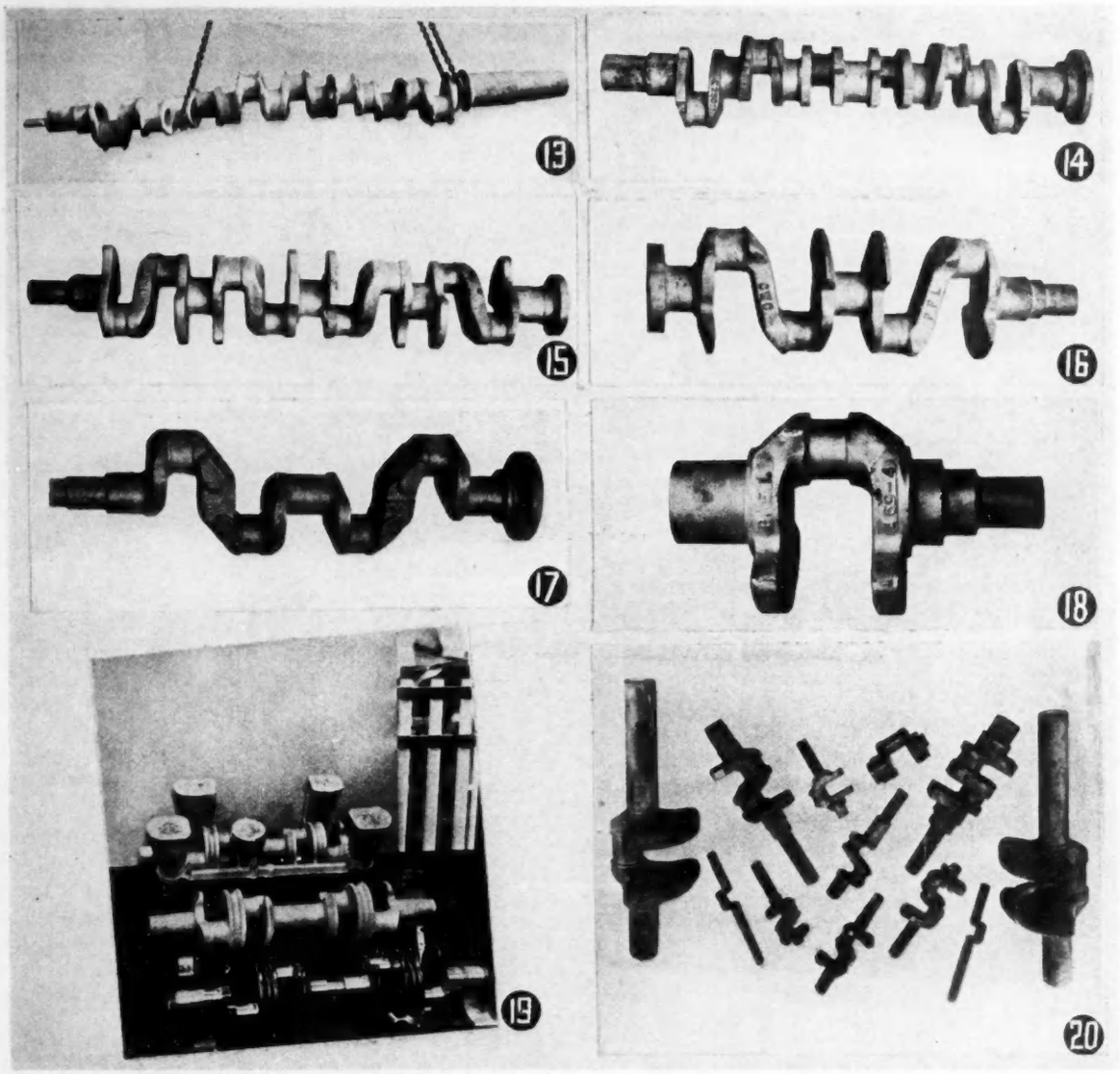
Proferall 3X	
Total Carbon	2.40-2.80
Silicon	2.25-2.75
Sulphur	0.10 maximum
Phosphorus	0.15 maximum
Manganese	0.80-1.20
Nickel	1.00-1.20
Molybdenum	1.00-1.20
Tensile Strength	60,000 to 80,000 lb. per sq. in.
Brinell Hardness	265 to 320 as cast

No heat-treatment is required, and the same advantages may be claimed as listed under steel. The large Diesel shaft, Fig. 13, weighs 1920 lb. as cast, whereas a forging weighed approximately 3600 lb. It is not possible to compare the finished weights, as a forging could not be machined to the extent to which the casting has been cored out.

The Caterpillar Tractor Co. was the first to adopt a nickel-molybdenum gray-cast-iron crankshaft as a standard part for their two-cylinder starting gas engine. This crankshaft is unique in that it represents the production of high-strength iron direct from the cupola using high steel mixes and graphitizing in the ladle. With several years of successful service in the background it is proved that by proper foundry control reliable castings are a certainty. Fig. 19 shows methods of gating and machining as developed by Caterpillar. Composition and properties of these crankshafts follow:

Caterpillar Starting-Engine Crankshafts	
Total Carbon	2.95-3.05
Silicon	2.00-2.10
Nickel	1.50-1.60
Molybdenum	0.75-0.85
Manganese	0.70-0.80
Sulphur	0.06-0.08
Phosphorus	0.05-0.07
Tensile Strength	50,000, lb. per sq. in.
Brinell Hardness No.	260-280

An illustration of the wide use to which alloyed-gray-iron crankshafts have advanced may be gained from Fig. 20. Al-



Figs. 13 to 18 Courtesy Campbell, Wyant & Cannon Foundry Co.
Fig. 19 Courtesy Caterpillar Tractor Co.

Fig 20 Courtesy Frank Foundries

Fig. 13 – Diesel Crankshaft; Weight 1920 Lb.

Fig. 14 – Diesel Crankshaft – Six Cylinder

Fig. 15 – Crankshaft of Eight-Cylinder Automotive Engine

Fig. 16 – Crankshaft for Four-Cylinder Gas Engine – Weight, 40 Lb.

Fig. 17 – Crankshaft for Four-Cylinder Gas Engine; Weight, 40 Lb.

Fig. 18 – Single-Cylinder Heavy-Duty Crankshaft, Weight 80 Lb.

Fig. 19 – Two-Cylinder Starting-Engine Crankshafts

Fig. 20 – Frank Foundries Crankshafts

though these applications are relatively small in size when compared to the size of a Diesel crankshaft, nevertheless they are of relative importance from the standpoint of dependability. These castings were made by Frank Foundries Corp. and are known as "Frankite"; they are of the composition given in the opposite table.

In addition to the several compositions already mentioned, there are a great many others, such as "Meehanite," "Gunite," "Gun Iron," and so on. The producers of these trade-marked castings are to be complimented on the splendid job they have done in advancing the "art" of founding to the science of casting.

In conclusion, I wish to express my appreciation to the men of the companies already referred to and to Messrs. Eash, Wood, and Hergenroether of The International Nickel Co., Inc., for their assistance in the preparation of this paper.

Frankite

	Large	Small
Total Carbon	2.75	3.00
Silicon	1.90	2.30
Sulphur	0.08 maximum	
Phosphorus	0.20 maximum	
Manganese	0.75	0.55
Nickel	1.50	1.00
Molybdenum	0.75	0.50
Tensile strength	50,000 lb. per sq. in., minimum	
Brinell Hardness No.	230-280	

Metal duplexed (cupola and electric furnace)

The Aircraft Trend in Body Design from the Aircraft Viewpoint

By Ralph H. Upson
Consulting Aeronautical Engineer

AIRCRAFT have taken more from automobile design than they have given, but they can now repay much of the obligation without necessarily transgressing the requirements of production economy and reasonable design stability. Some of these possibilities are:

(1) Improved streamlining of necessary exposed parts, particularly underneath, and incorporation of other accessories in the general body lines.

(2) Use of curved glass in the windshield and lightening of all window material.

(3) Reduction of the frame to the status of an assembly unit, with structural significance only in combination with the body.

(4) More effective distribution of flange material around the doors.

(5) Lightening of skin by use of internal stiffeners, particularly on top.

(6) Development of a smaller, more efficient radiator and lightening of various engine parts.

Most important is the mental attitude behind the work. Phenomenal production economy has been attained because almost every man in the industry has concentrated on it. Worthwhile weight and drag reduction would result if even one competent man were assigned specifically to that duty.

Lightening, with streamlining, is a beneficent cycle that permits still further lightening, resulting directly in substantial operating economies and, indirectly, in lower production cost.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937.]

FROM the aircraft viewpoint any present discussion of the aircraft trend in automobile design must be mainly in consideration of future possibilities because, to date, such a trend has been more apparent than real – and more imagined than apparent.

Aircraft have much to thank the automobile for: the gasoline engine; wheels, tires, and brakes; safety glass; door and window hardware; interior trim; advances in steel and other materials; improved fuel and oil. None of these things, however, were copied by the aircraft engineer. Caring nothing for the appearance and, it must be said, but little for the cost, he adapted to his purpose the substance rather than the semblance of such developments as he found them usable.

The present so-called aircraft trend in automobiles is mainly of quite the opposite kind. Aircraft have caught popular fancy but, in spite of the psychological urge to fly, it is still not usually feasible for an individual to buy an airplane. Instead, he must be satisfied with an automobile dressed up like an airplane, in the same way that perhaps earlier in life he "fell for" a bright red "Fire-Chief" velocipede.

Use of Streamlining

On some such supposition, streamlining, the most obvious refinement about aircraft, is turned over to the automobile stylist who in most cases takes the semblance and lets the substance go – and even at that, it takes a good deal of imagination to see much airplane design in a present-day automobile. Practical results are even more lacking because, in spite of some real benefit from the smoother profile lines, the net gain is but a small fraction of what is readily possible. Specifically, the basic trouble is failure to secure a balanced design. As the direct object of streamlining is to cause the air to close in behind without permanent displacement, there is little to be gained by "streamlining" the top alone; similarly, streamlining the front is ineffective if the air concerned is to be caught by projections and "dead spots" further back; and fine lines at the rear have no chance to act unless the air can be led smoothly to them.

In the last analysis, a proper balance and suitability for the purpose is the principal basis for good looks; it is certainly the basis for the eye appeal of a modern airplane; and for lack of such basis most of the lines affected by present automobiles cannot long endure, either practically or artistically.

In the matter of weight, we again have the aircraft trend, psychologically speaking. Only a few years ago it was common for manufacturers and dealers to boast about the amount of weight in their cars. Now the tendency is more the other way;

we begin to hear about the value of lightness and an "airplane type of structure"; but still the practical results are negligible. Again perhaps the main trouble is a lack of balance or unity in the design.

The traditional pound basis of figuring cost, although perfectly legitimate as an accounting expedient, is apparently a bad influence on any serious attempt at weight saving. Although many materials may be most conveniently purchased on a pound basis, there is apparently not the slightest excuse for thinking of the finished car that way. To take an example in round numbers: suppose a car priced at $850 weighs 3150 lb. empty, and 4000 lb. fully loaded. Is the purchaser buying 3150 lb. of *car* at $0.27 per lb.? Rather is he not buying 850 lb. of *carrying capacity* at $1 per lb.?

It must be granted of course that the last 200 lb. of carrying capacity are not so important as the first 200. Also there are other things which the purchaser is buying in a car, such as space, performance, comfort, and so on; but he is certainly *not* interested in pounds of car, unless perhaps he is still hanging on to the old myth that a heavier car makes riding easier. If he is interested in improved economy rather than increased carrying capacity, weight saving directly affects the major part of his operating expense, including fuel, engine, and spring depreciation, brake bands, and tire wear, especially if the lightening is combined with a similar proportion of drag saving.

From the manufacturer's viewpoint, every direct saving of weight permits more saving if advantage is taken of the allowable incidental reduction in power, wheel and spring material, fuel capacity, and general stiffness and strength. Again assuming proportional reduction of air drag, for every 2 lb. initial saving, it appears possible on the average to save at least 1 lb. more.

Weight Saving in Aircraft

This saving cannot be made by copying airplane details, any more than a successful airplane can be built from present automobile parts. On the other hand, aeronautical engineering, as it stands today, is a veritable mine of ideas and inspiration on weight saving of all kinds, many of which certainly can be adapted to automobiles with little if any penalty in cost and with no sudden disturbance of traditional general arrangement. In the end, it is believed confidently that the genius of the automobile engineer for reducing costs will reassert itself by taking economic advantage of the generally reduced weight. This action is to be expected especially where the weight saving requires no great increase in the cost of material, as in the usual case where stiffness, rather than strength, is the main criterion.

In respect to lightening automobile construction, the advantage possessed by the aeronautical engineer is not his familiarity with aircraft, but the fact that necessity has compelled him to use his ingenuity in hundreds of different ways to minimize weight in almost every move that he makes. As an example of the results achieved, take one of the better airplanes carrying a useful load that is about the same (900 lb.) as the automobile already mentioned, but providing for extra fuel instead of the fifth passenger. The airplane weighs considerably less than half of the automobile weight, in spite of wings, tail, and an engine of about twice the horsepower. For the parts roughly corresponding to the various automobile components, the approximate weight would be more like one-fourth that of the automobile.

Almost needless to say, the cabin is a unit structure, rather than a body separate from the frame. A similar principle of construction should be adopted for automobiles, not because it is used for airplanes but because it is rational and sensible, thus providing one of the many possible means of weight saving by basic arrangement of parts rather than by costly detail refinement.

In one way the conventional frame sounds logical enough. The present body, by itself, is rigid but lacks strength. Combined with a strong, although flexible frame, it might be supposed off-hand that a combination of strength and stiffness would result. Actually it is a good deal like trying to combine a stiff paper cylinder with a strong piece of rope: in any such direct combination in parallel, the stiffness, of course, will be destroyed by the time the full strength is realized; or alternatively, the increase in strength will be negligible if the stiffness is to be retained.

A shallow frame and a deep body naturally fight each other – they are absolutely incapable of real cooperation. The resultant shortcomings have been met by various relief measures, such as interposing rubber between the two elements to absorb a greater relative movement; avoidance of joints that will creak and rattle; redesign of the frame to increase its own stiffness; and, at the same time, strength has been added separately to the body for protection of the occupants. Why all this complication when a simple basic remedy is at hand? In brief, the issue is apparently not whether to adopt some aircraft trend, but whether a car should not be designed as a rational unit, instead of as a collection of unrelated parts.

Specific Possibilities

Some of the specific possibilities for reduction of drag and weight are:

(1) Improved streamlining of necessary exposed parts, particularly underneath, and incorporation of other accessories in the general body lines.

(2) Use of curved glass in the windshield and lightening of all window material.

(3) Reduction of the frame to the status of an assembly unit, with structural significance only in combination with the body.

(4) More effective distribution of flange material around the doors.

(5) Lightening of skin by use of internal stiffeners, particularly on top.

(6) Development of a smaller, more efficient radiator and lightening of various engine parts.

Most important is the mental attitude behind the work. Phenomenal production economy has been attained because almost every man in the industry has concentrated on it. Worthwhile weight and drag reduction would result if even one competent man were assigned specifically to that duty.

Lightening, with streamlining, is a beneficent cycle that permits still further lightening, resulting directly in substantial operating economies, and indirectly in lower production cost.

The paper "Aircraft-Engine Installation Vibration Problems" by John M. Tyler, printed in the June, 1937, S.A.E. JOURNAL, was originally prepared by Mr. Tyler in the fall of 1936, while he was a member of the engineering staff of the Lycoming Division of the Aviation Manufacturing Corp. He joined Hamilton Standard Propellers Division of United Aircraft Corp. in April, 1937.

Die Castings for Automotive Parts

By C. R. Maxon
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ADAPTABILITY of die castings to economical manufacture, to close dimensions, to complex shapes, and to thin sections is stressed.

Die-casting alloys are reviewed, emphasizing the suitability of the zinc alloys in cost, physical properties, and die cost, along with their limitations as to temperature and otherwise.

A discussion of applications to automotive grilles includes comparison with other methods of producing these parts and a description of the built-up type of die-cast grille.

Steering-wheel applications considered include hubs, horn rings, and light switches. Windshield-frame parts, louvres, radio grilles, lamp mountings, body-interior parts, horn bodies, and a number of chassis applications also are discussed.

A review of finishing processes and notes and suggestions on design conclude the paper.

THAT die castings possess exceptional utility for a wide range of automotive parts is too well recognized to require emphasis here. Among the chief reasons for their extensive use are:

(1) Adaptability to rapid production at moderate cost and within closer dimensional limits, not only as compared to other castings, but also as against stampings and forgings.

(2) Adaptability to a wide range of shapes and sizes, including complex parts with difficult core work – irregular contours and structural elements which, though not so strong, of course, as wrought parts, have unusual toughness (high impact strength) as compared to most low-cost castings.

(3) Producibility in some instances with surfaces so smooth as to require little or no polishing or even buffing prior to plating or other applied finishes.

(4) Producibility in remarkably thin sections, yet with a stiffness not attained, as a rule, with stamped parts – and often with the metal disposed (as in bosses and ribs) in a manner not feasible in stamping and forging.

(5) Relative ease of machining, when machining is required, and with a minimum waste in metal removed.

(6) Comparative resistance to corrosion under most conditions of service, such corrosion as does occur being confined to the surface and rarely affecting the strength or serviceability of the part.

There are, of course, many other reasons for the use of die castings, as will appear in the discussion of specific applications hereafter. There are also certain limitations which must be observed, as with other castings and indeed with all automotive materials. It is well to remember, however, that the utility of the die casting is far from being confined to decorative parts. Many structural elements are used, as later shown, and there is likelihood of further developments along this line.

Other papers presented before the Society have gone quite fully into the die-casting alloys available. The properties are covered in detail in the booklet, "Zamak Alloys for Zinc Alloy Die Castings." Hence it is not necessary to detail this phase of die castings here. Zinc alloys account for over 95 per cent of automotive die castings and, hence, are considered almost exclusively here. Some die castings based on aluminum, copper, magnesium, lead, and tin, are or have been used chiefly for rather special applications in the automotive field.

Besides the low cost of the zinc alloys themselves, factors in their favor include: (1) generally excellent physical properties; (2) low die cost, because the metal casts at low temperature and dies do not require heat-treatment. Because of these advantages, some die casters employ zinc alloys exclusively and almost all serving the automotive industry confine all or nearly all their automotive production to zinc alloys.

Limitations of zinc alloys include those involving temperature in service. Prolonged use at temperatures above 300 deg. fahr. is not recommended. Impact strength is high, as already indicated, under normal temperatures, but decreases rapidly at very low temperatures, although the impact strength is regained with the return to normal temperatures. Despite this fact, breakage attributable to it is rarely reported. It should be kept in mind by designers, however, and an adequate factor of safety allowed, especially with structural parts. With S.A.E. No. 921, containing 2.7 per cent copper, the high initial impact strength is reduced to about one-third this value by three years of normal aging in service. The S.A.E. No. 903 alloy, which is copper free and now very widely used, has equally high initial impact strength but does not decrease under any aging condition, although it is temporarily reduced by low temperature, as already indicated.

A somewhat newer alloy, No. 925, has been recently adopted by S.A.E. It is the equal of No. 903 in permanence of dimensions and impact strength whereas, at elevated temperatures, growth of dimensions and loss of impact strength are of lesser magnitude and slower rate than those of No. 921. It may be used in virtually any application for which No. 921 is suited and, being somewhat more resistant to surface corrosion than No. 903, may be used when additional corrosion resistance beyond that of No. 903 is required or when better permanence than can be obtained with No. 921 is desired.

The foregoing remarks apply, of course, to standard specification zinc alloys made from high purity (99.99+ per cent) zinc and in which the tin, lead, and cadmium content is held rigidly below the maximum allowable limits. All reliable die casters adhere closely to standard specifications and many of them have recently installed modern spectrographic equip-

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ment for rigid checking. Alloys made to standard specifications are entirely free from the destructive intergranular corrosion which many years ago was encountered with die castings in which good alloying practice was not followed.

As with most metals, exposure to moisture, especially at elevated temperatures, tends to increase surface corrosion. Continuous or prolonged use in contact with hot water or steam is not recommended. Normal exposures in service of automotive zinc-alloy die castings seldom result in corrosion difficulties, but when they may do so, as in occasional carburetor castings exposed to fuels containing water, a simple dipping treatment in a suitable solution affords adequate protection. This treatment, known as "Cronak," is patented by The New Jersey Zinc Co.

Applications of Die Castings

Just as "the proper study of mankind is man," so the proper study of die castings affords the best insight into their reason for being, and should help to bring out many facts helpful to designers and other engineers interested in future applications. The following examples are selected either as being typical of good current or recent practice, or because they embody certain more or less "special" features deserving of consideration whenever new designs are up for discussion. In presenting these examples the author acknowledges indebtedness to many automotive and die-casting engineers who have contributed liberally from their experience for the benefit of the S.A.E. and its members. Many of the comments made, however, are a result of the author's own rather extensive observations concerning die castings.

It is inevitable in any paper that comparisons be made of various materials and fabricating methods. The writer has attempted to present competitive materials in as fair a light as possible.

Grilles

Although grilles may be regarded as a somewhat special form of die casting, they are prominent because of their position on the car. They are die cast, in fact, quite largely because of their prominence and because it is important from a sales standpoint to have a grille of fine appearance. Since appearance is a matter of individual preference and preferences differ, besides being influenced in some cases by commercial considerations, there is undoubtedly a difference of opinion as to whether die-cast grilles are better looking than the stamped variety, but we think it is conceded that exceedingly fine appearance has been attained in the die-cast grille. The author leans to the view that the die-cast grille is invariably better in appearance as well as better in other respects than commercially feasible stamped types, although it is not always the cheapest, and perhaps never the lightest, construction. If this view is correct, it may well be asked why fewer cars use the die-cast form in 1937 than in 1936 models. Many factors enter into this consideration. One of the reasons for writing this paper is to answer this question, so far as is feasible.

An important and perhaps the primary reason for dropping die-cast grilles on some 1937 models was that production facilities were overtaxed in supplying some 1936 models, with consequent delays in deliveries. They were overtaxed largely because the number of die-casting machines then available for making one-piece grilles was limited in proportion to the demand. Polishing and plating facilities also were heavily taxed.

Today there are much better machines available. Their number is not great, but they are simple and effective and more could be built if orders for 1938 grilles warrant, and they are placed early enough. The placing of orders at the eleventh hour multiplies difficulties. It is also likely to result in grilles which are both heavier and more expensive than they need be if proper forethought were given to the early placing of orders. A sufficient allowance of time should be made for the die caster to try out a die for thin sections before going to heavier ones, if they are needed, to get a good job. At least one grille in production today is, in the opinion of one of the most competent die casters, fully 20 per cent heavier than it need have been if he had been given a little more time to make and try out the die. Deliveries were required so soon after placing the order that sections had to be made such as he knew would work. Had more time been allowed he would have tried and probably succeeded with thinner sections, but there was no time to experiment. It is easier, comparatively, to increase sections when they are found too thin, of course, than to attempt to reduce them once the die is built.

Some grilles are lighter this year than last and others could be made lighter than they are by (1) greater forethought in design, (2) using the grille as a supported rather than as a supporting unit, (3) making the grille of smaller size and/or with larger openings, and (4) using a design which permits of thinner sections. Some of these possibilities have scarcely been thought of, let alone being given adequate consideration. They might go far toward eliminating the criticism of undue weight, which now is a just one. Although it is doubtful if the weight can ever be reduced to that of the stamped grille, the die-cast grille gives both the appearance and reality of stiffness and sturdiness. However this consideration may be regarded from a strictly engineering angle, it is being turned to effect, perhaps a little unfairly, in selling when the prospect is asked to compare the "feel" of a die-cast grille with the decidedly flexible sheet-metal type on cars in the same price class.

Another possibility in die-cast grille construction yet to be fully explored is that of the built-up type, of which the Oldsmobile-6 grille is this year's noteworthy example. See Fig. 1 (*right-center*). We understand that an important consideration in going to a built-up design had to do with die construction. Four dies are used for making the component parts and, as more men could work on these than on a die for a one-piece job, the dies could be built quicker. Though not a factor in this instance, these dies can be and are used in smaller machines than are required for one-piece grilles, and many more of such machines are available. In general, the smaller the casting, the thinner it can be made, and this construction may result in weight saving, although we understand there was none in this instance. In this grille the castings are of nearly semicircular shape and they are placed in pairs on a rotating fixture, which facilitates polishing in an automatic machine.

It is understood that this grille was suitable for either built-up or one-piece construction, but it is difficult to see how it could have been polished with satisfaction in one piece. In any case, the polishing set-up is efficient, and production cost is reported somewhat lower than for a one-piece design, especially with respect to plating. An advantage of a built-up grille is that it is not necessary to replace the entire unit if an accident results in breakage or injury of one or more parts while others remain suitable for use. The Oldsmobile-6 grille is serviced in parts. A complete assembly lists at $15, and the major parts at $1.75 and $1.90 each. Dealers and users are reported as well satisfied with this arrangement, and a construction of this kind should go far toward avoiding the criticism of high replacement costs leveled at the die-cast grille. Such costs have been too high in some cases, partly because a mark-up of about 4 to 1 has been in effect.[1] Much lower mark-ups are now charged in some instances, and labor charges have been reduced by making fastenings readily accessible. Breakage is usually the result of another car backing into a grille or of ramming the car ahead with the grille. Such accidents can be minimized by improved bumper design or by the use of inexpensive supplementary guards.

[1] This statement has been confirmed by members of the industry.

Such changes, and perhaps the use of grilles which are V-shaped or well rounded as viewed from above, and with the stronger bars of the grille horizontal rather than vertical, should go far to reduce breakage and resulting service costs and to make them negligible factors, as some contend that they are already. Of course no grille, stamped or die cast, can be rendered proof against accident, but the designer should give consideration to methods of preventing damage by minor accidents.

There is much to be said in favor of constructions on the order of that used in the 1937 Buick grille made in two die-cast sections with a stamped framework and center section, although it would seem that the die-cast parts might be lightened. See Fig. 1 (*left*). Other combinations of stamped and die-cast parts may well work out to excellent advantage. Sections with short horizontal bars can be cast and polished readily and can be replaced separately if injured.

Today, those who style the car appear to have most to say as to the design of grilles and of other non-functional members to be die cast. Their first concern is naturally and properly with appearance, but there are undoubtedly many cases in which they could save in the expense as well as in the weight of die-cast parts if they would work closely with the die caster in the early stages of the design, and without any sacrifice in appearance of the finished design. Three 1937 cars have die-cast grilles of honeycomb design. These grilles are difficult to cast with a good finish suitable for plating and are also hard to clean of fins. Each hole, of which there are over 1200 in some designs, means a core and often a resulting fin which is almost sure to require some hand-filing on four sides of each hole! Machine-cleaning is done to some extent where a saving results, but some hand-cleaning is unavoidable, and the cost is high when so many holes are involved.

In respect to costs, that for the die and cleaning tools for a one-piece die-cast grille usually runs from $14,000 upward, perhaps to $25,000 or more, but is far below the tooling for a stamped grille and may bring the total cost per grille, tools included, lower for the die-cast than for the stamped grille, at least where the total number of grilles required is moderate. Upward of $300,000 is said to have been the tooling cost for making a stamped grille for one of the highest production cars. The production rate is high and could not be equaled with a single or perhaps even two or three die casting dies, but the difference in tooling is striking even if offset by lower unit costs, especially when the difference in appearance of the product is considered.

Comparisons of this sort may prove misleading, especially as the number of factors involved and the differences in results required may be great. It hardly can be doubted, however, that the die-cast grille has made a deserved place for itself in car design and is likely to see wider rather than restricted use if designers make the most of its possibilities.

An engineer with one of the most experienced users of die-cast grilles recently wrote as follows: "In our opinion, die castings will increase in usage in the automotive field, particularly as applied to body and sheet-metal parts. In fact, die-cast parts are on the increase because of their neatness of appearance, accuracy and uniformity of dimensions, also their cheapness as compared to stampings of equal appearance. We realize that many die castings may be replaced with stampings but, in most instances, the part suffers greatly in loss of character in adapting it to a stamping.

"We do not consider die-cast grilles more expensive than other types which have the same fine characteristics and appearance. When we adopted the die-cast grille in 1935, the most important reason for doing so was based on the cost standpoint. During the previous year, our grille costs were nearly 50 per cent higher than the cost of the die-cast grille, due principally to inability to control accurately the stamped parts which made up our grille and to control accurately the size and shape of the assembled stamped grille. Die-cast grilles have one great advantage, that is, dimensions, shapes, and appearance may be held uniform. We do believe that the die-cast grille has presented a sales asset."

Steering-Wheel Applications

Die-cast hubs for steering wheels, especially the deluxe type with wire spokes, are extensively used and are an excellent example of an application which is largely structural in character. Fig. 2 gives typical examples. These hubs vary in weight from about 1.3 lb. to about 6.5 lb., a difference which would seem to indicate that some hubs are heavier than they need be for purely structural reasons. Differences in size, which in turn is dictated in part by the stylist's ideas of correct proportions, have much to do with the weight but, where weight saving is considered important, a study looking toward better metal distribution might well be made in some cases.

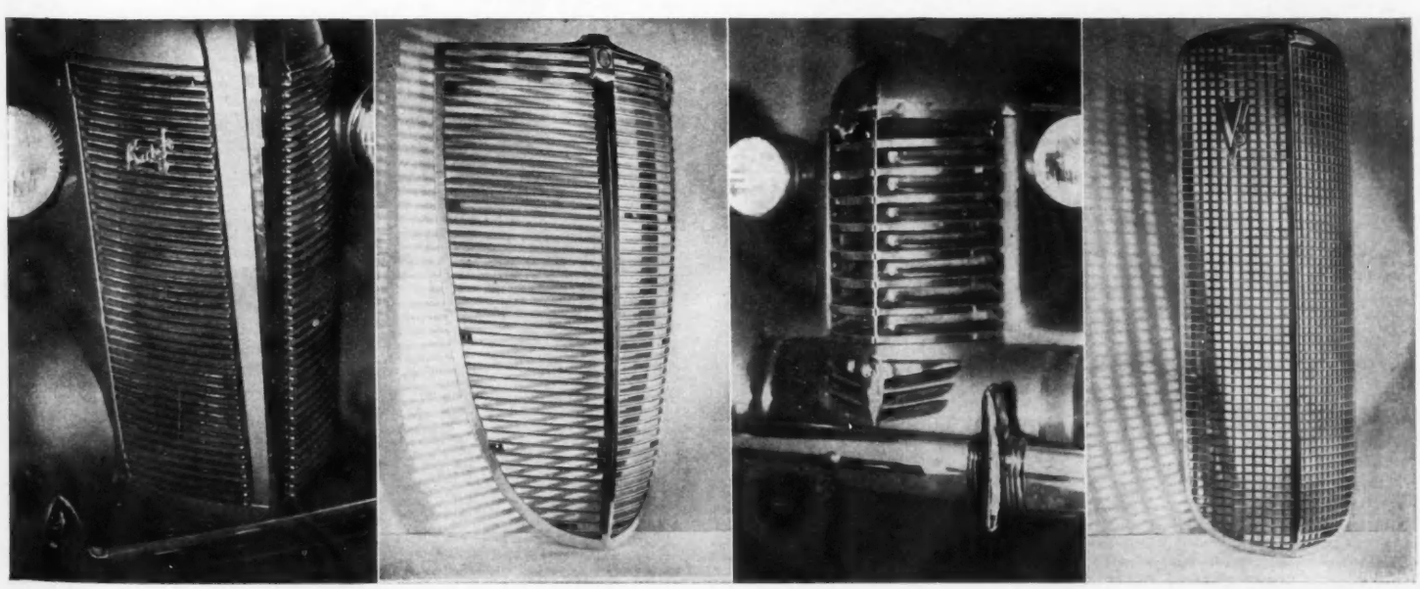

Fig. 1 – Die-Cast Grilles

It is quite common practice to use a steel insert to which spokes are welded, and it might result in some weight saving if the insert were extended to include the keyway, where stresses may be highest.

Differences in the design of interior parts of the hub are chiefly those incident to spoke fastening. One of three methods of fastening is usually employed: welding to a steel insert; use of a ring fitting into an annular groove which includes notches cut in the spoke; or screwing the spokes into place. In some designs using a steel insert, the latter is put in place after casting. This method saves some time in the casting cycle and has the advantage that welding surfaces are free of zinc. Where the insert is cast in place it is, in at least one case, made a press fit on the core pin and has to be forced over this pin before the die is closed for casting. This operation requires some special tooling and lengthens the casting cycle with no apparent benefit except to keep the zinc away from the welding surface.

Three to five wires are used to form each spoke and the holes for these can be cored if the axes of those in each group are made parallel rather than radial. If flat-strip spokes displace some round wire, as in the Olds wheel, it is necessary to core the holes for them because of the expense of machining a hole to fit. The coring of holes for wires of circular section as well as those for flat strip naturally increases die cost and may lengthen the casting cycle slightly, but it avoids the need for drilling. A good set-up for drilling using, for example, three Kingsbury or equivalent drill units and an indexing fixture permits of rapid drilling at low cost, even though the drilling of small deep holes requires some care to avoid drill breakage. When spokes are threaded into the holes, it is possible to make them form their own thread, hence separate tapping is not essential.

Despite the stresses imposed on steering-wheel hubs, the author does not know of any cases in which breakage has occurred. Fig. 3 shows what happened in the wreck of a Buick "60" in a collision resulting from the car running down hill out of driver control and striking a street car head on. The driver, thrown against the wheel, was hurled into the rear seat sustaining a broken leg and minor head injuries. Although the steering column is bent and the wheel was forced forward so that it dented the instrument panel and was badly distorted, neither the spokes (of Firth Sterling stainless steel) nor the die-cast hub were broken, in spite of the fact that the engine was broken loose and pushed back to within about 18 in. of the driver's seat. Had the wheel broken, the driver would probably have been much more seriously injured if not killed.

Fig. 3 – Interior of a Wrecked Buick "60" – The De-Luxe Spring Spoked Wheel Is Bent but not Broken and Probably Saved the Life of the Driver

This incident appears to indicate that a flexible-spoke steering wheel may be a safety factor besides having other advantages.

Other applications of die castings to the steering wheel include the horn ring as shown in Fig. 4 on the 1937 Buicks with de luxe steering wheels. This ring, spokes included, is a light and surprisingly flexible one-piece die casting of zinc alloy. It is cast with the hub web solid and this hub is afterward split by saw cuts equi-distant between spokes.

The horn ring is also a safety feature as the driver need not remove either hand from the wheel to sound the horn.

Ford light-switches, which surround the horn button, are being die-cast this year, whereas a year ago they were molded from plastic and had a die-cast insert. This change is understood to have represented some cost saving and also to have avoided certain difficulties in color matching encountered with the plastic. A Macoid finish to match other interior parts is now applied. Castings are made in a two-cavity die running

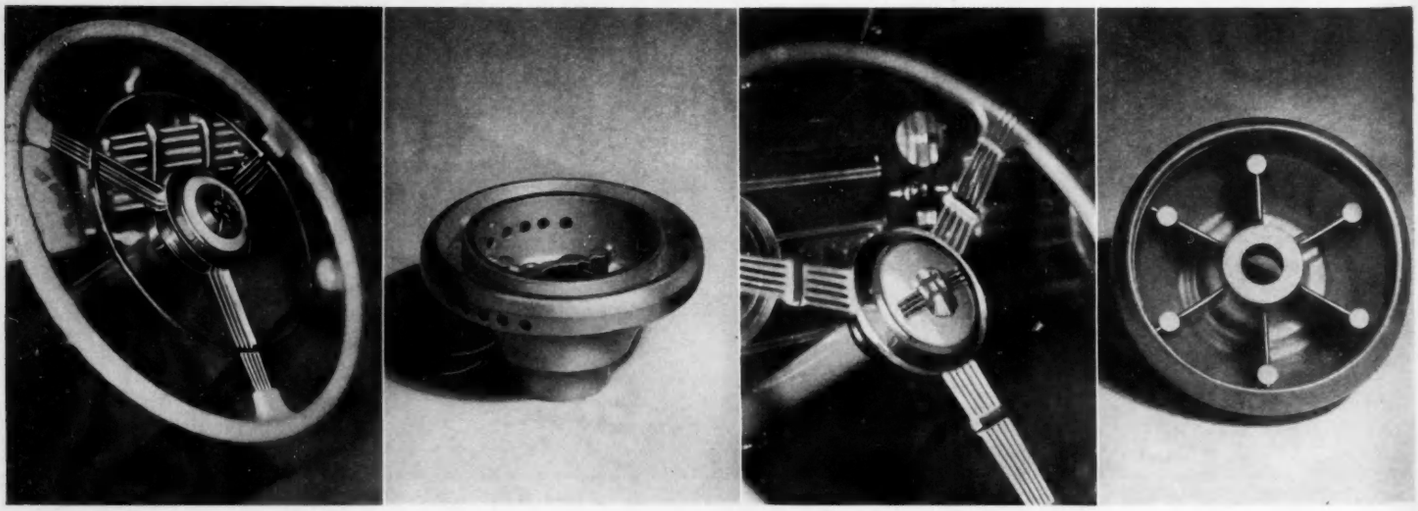

Fig. 2 – Typical Examples of Die-Cast Steering-Wheel Hubs

400 to 450 shots per hr., and giving a very smooth surface. The only subsequent operation on these castings prior to applying the finish (aside from breaking off the gate and inspection) is an operation performed in a punch press at the rate of about 1500 pieces per hr.

As against this record, is that of a steering-wheel hub cast in a two-cavity die at about 100 shots per hr., the slower rate being a result of using a steel insert which has to be forced over a core pin (to prevent zinc from coating the inner surface). A special loading fixture for the inserts is needed and has to be lowered between the die halves between shots and the die partly closed to force the rings over the core pin. Cleaning of the hub is done in a press with a shaving die after which the hub is inspected and shipped. Spoke holes in this hub are drilled subsequently in the car manufacturer's plant, where also the central hub hole is reamed and the keyway broached, spokes inserted and welded to the steel insert. Finish includes a priming coat baked for 15 min. and a final coat baked 2 hr. and 40 min.

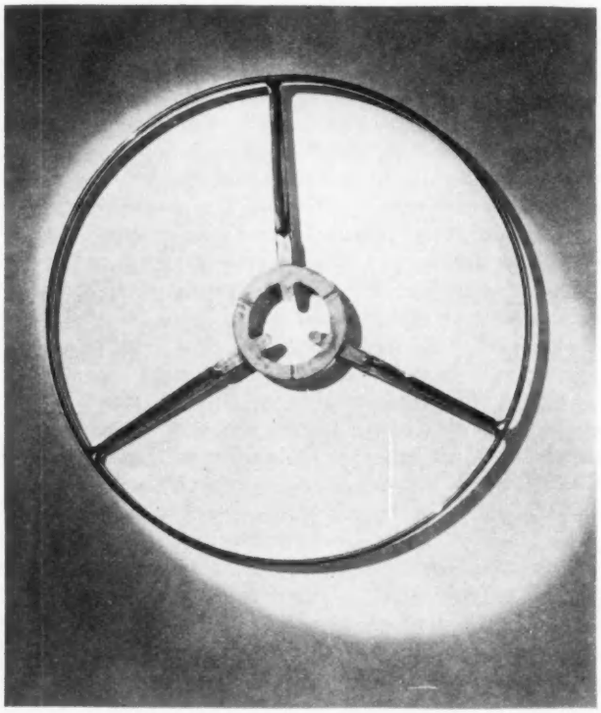

Fig. 4 – Horn Ring for Buick Steering Wheel, Die-Cast in One Piece and Afterward Cut Apart at the Central Ring

Windshield and Window Frames

The use of die castings in windshields for open and convertible models is far from new. Some complete windshield frames have been die-cast in one piece, one Cadillac job weighing 33 lb. and measuring 50 x 22 in. overall, being among the largest die castings produced. Only 400 to 500 a year were required, yet the saving in finishing as compared to a sand casting warranted the constructing of the very large die. This frame was used some years ago and we understand that it is not now in production. With the introduction of improved methods of body construction, including improved welding practice, windshield design has changed radically.

Many windshield stanchions and separate windshield frame parts are still used and serve their purpose well, but such stanchions are exceedingly heavy in most designs with which the author is familiar. Die castings are used for these parts primarily because they supply the required sectional contours and yield a non-rusting support which is readily plated or otherwise finished. It is the author's opinion that weight might be decreased greatly and strength increased by using a properly designed steel insert running from end to end and thinner sections, although this change might not decrease cost. As present designs, in general, are used only on models in which production is relatively small, studies of weight saving appear not to have gone as far as they might. Certainly, if stanchions which are more slender were provided, the blind spot which they create could be reduced with some gain in safety.

Another example of structural use of die castings is in the frames for D-shaped ventilating wings and quarter windows. A good example of the latter application is the Packard frame shown with its rubber cover in Fig. 5. Such parts have the required supporting bosses for pivots and limit stops and can be made quite light, although perhaps a little heavier than formed steel frames. Important advantages are the closer dimensions which can be held and the freedom from rusting which the use of a zinc alloy secures. When a rubber cover for sealing purposes is desired, the die casting can be brass-plated and used as an insert in the mold forming the rubber cover which is vulcanized to it.

In some General Motors applications of this nature, the outer frame for a front window has been die-cast, roughly in an F-shape as shown in Fig. 6. The lower projection extends into the door and the center section is adapted for mounting a die-cast regulator having a vertical pivot about which the inner frame, carrying the glass, turns. This makes a substantial superstructure as well as a firm fastening for screwing or riveting to the frame of the door itself. The assembly takes the place of one which, if not die cast, would have to be built up from several stampings which would probably cost more and be much more difficult to hold within the desired dimensional tolerances.

Although not properly classified under the heading of frames, mention is made here of some closely associated parts

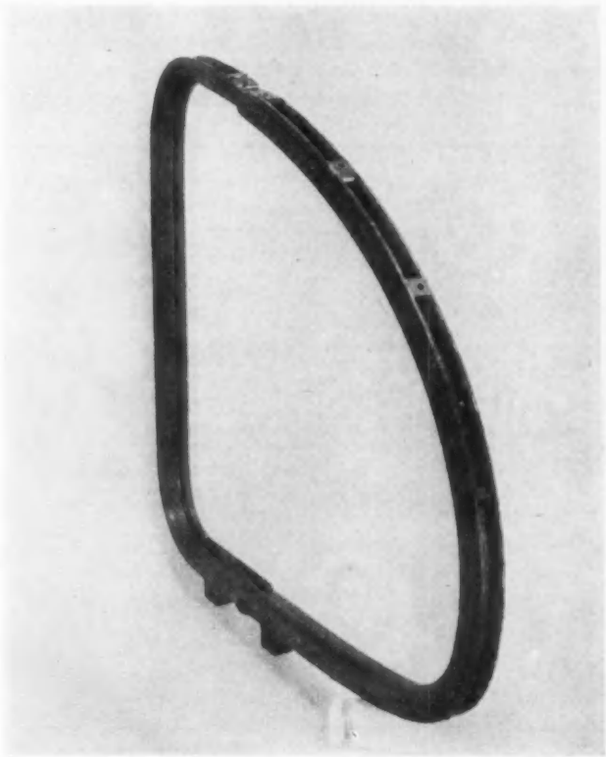

Fig. 5 – D-Shaped Frame for Packard Ventilating Quarter Windows, Die-Cast with Pivot Hole and Stops – Subsequently the Die Casting Is Brass-Plated and Used as an Insert around which Rubber Is Vulcanized

of doors, including window regulators, lock parts and dovetail assemblies. Many dovetail bumpers have been die-cast, and the striker plate for holding these is now being die-cast also, and is understood to be standing up well in service despite the shocks to which it is subjected. Such castings are simple to make and easier to hold within close dimensions than stamped or sand-cast parts. One large maker of door locks who heretofor has always stamped the lock frame, is now experimenting with a die-cast shell, again because of difficulty in holding the desired tolerances in the corresponding stamped part. His only question concerning the die casting is whether it has enough resistance to shock in extremely cold weather, and there is sufficient expectancy of success to warrant his building a die and putting out some experimental locks for trial. Lock cylinders and other small lock parts have been die-cast by the million and are in use wherever a lock is required in the modern car. Intricate parts of this kind are more or less ideal for die-casting and cannot be duplicated with accuracy and faithfulness to detail, at comparable cost, by any other means.

As already noted, window regulator parts, which may include gears and sprockets as well as housing parts, are readily die-cast and give good service in that form, frequently being lower in cost than equivalent stamped or machined parts. Hardware for regulators and doors is treated under another heading.

Louvres and Exterior Body Parts

Extensive application of die castings as louvres and decorative applications on hoods is a further tribute to their fine appearance and low comparative cost. Again it must be admitted that the die casting in such applications adds weight which is justified chiefly on the basis of improved appearance – and appearance which cannot be attained at all with stamped parts in some cases, and in other instances only with a much larger expenditure in tooling. Those interested in weight saving can undoubtedly gain it and still take advantage of the die casting however, if they will set this objective, cooperate closely with the die caster, and give him an opportunity to work out die construction such as to yield the lightest feasible sections. Packard has accomplished much along this line in the louvre casting used on the new "6." This casting was made so light, in fact, that some breakage was encountered in the polishing operation, but a slight thickening of the section in one area quickly overcame this fault.

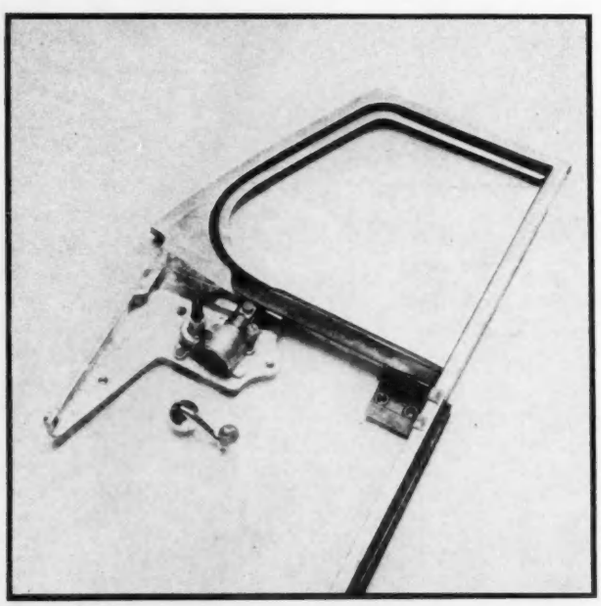

Fig. 6 – F-Shaped Die Casting for General Motors Car Having a Lower Extension for Attachment of Regulator Housing, also Die-Cast, and for Bolting to Door Frame – Dimensional Limits Are Held Much Closer than for Steel Frames

Incidentally, the casting in this case is used in close association with stamped louvres which carry through a somewhat similar motif as shown in Fig. 7, and the two die castings used are so light as to make at most only a few ounces difference in weight as compared to an all-stamped product. Variations of designs along this general line, in which die castings and stamped parts are used in close association, may well give many of the advantages of both types of parts and serve to decrease weight materially. Studies in this direction will pay both stylists and engineers. It is well to remember in this connection that it is entirely feasible today to die-cast either straight or curved moldings of almost any length up to 50 in. or more in a great variety of sectional shapes and surprisingly thin, but with convenient fastening bosses such as are not readily provided in stamped or rolled moldings. In some cases, these moldings can even be made flexible enough to conform to variations in stamped parts. An example of this is the D-shaped moldings, a pair of which are used to outline the grille on current Dodge models. These moldings have a section thickness of about 1/32 in. and are about 9/16 in. wide. Overall dimensions of the D-shaped casting are approximately 30 x 8 in. and represent about 60 linear in. of molding with bosses spaced about 5 in. apart. Casting is done at the rate of about 3 shots per min., and with so little flash that cleaning is a very simple and rapid process. The molding is quite flexible and conforms readily to inequalities of mating stamped parts.

As to irregularly shaped parts such as are used for decoration on and about the radiator shell and sometimes on fenders and headlamps, the die casting is so well suited as to have little or no competition. Such parts are usually designed largely by the stylist, who presumably leaves to the engineer the matter of fastening the casting to mating parts. Fastening is usually done by hidden screws or studs, some of the latter being cast in place in certain instances, but other methods of fastening are deserving of mention, as they may reduce the cost of the finished piece and of the assembly operation.

It is common practice to place studs or the axes of screw holes normal to the surface to which the casting is to be attached and, if this surface is curved as is usually the case, the axes of the studs or screw holes come at several odd angles. This arrangement often precludes the coring of more than one or two holes which may happen to come with axes normal to the die parting, unless the die is provided with separate slides which increase its cost and often are not feasible at all. The alternative for the die caster is to spot the holes, when the design permits, and to drill and usually tap them subsequently. This, of course, is a simple and not very expensive operation, but it usually involves shifting the casting through odd angles on the drill press. If the hole location and the angle of its axis are at all important, the use of a special rocking fixture is required which must be added to tooling cost. If all the holes were parallel, the fixture would cost less and all holes might be drilled simultaneously.

All this discussion points to possible advantages in certain cases of making some or all the holes parallel, and recessing or otherwise altering the sheet metal at attachment points so that parallel faces at bosses are presented to parallel boss faces in the die casting. Naturally, the cost of doing this altering would be set off against the extra cost of the operations on the die casting if it is not done, and the cheaper alternative selected. As the holes in the sheet metal are usually pierced in a separate operation anyway, it seems logical to suppose that some deformation in the sheet-metal surface might be made at the same time, in certain cases with little or no increase in the cost of that part. At present, it appears to be common practice

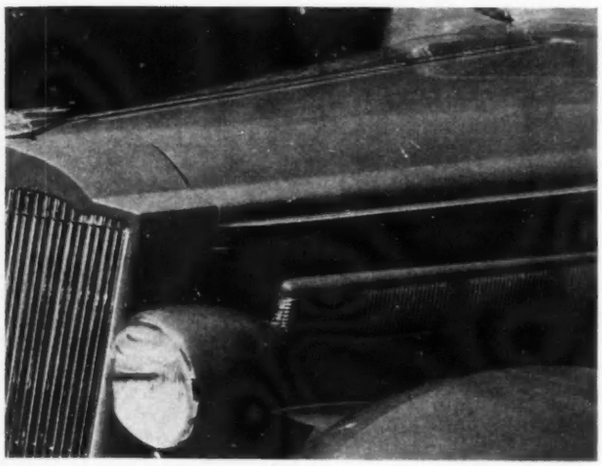

Fig. 7 – Hood of Packard-6, Showing Small and Very Light Die-Cast Louvres Associated with Harmonizing Stamped Louvre Openings

to give the "breaks" to the stamping shop and let the die caster worry as to fitting his product to the sheet metal. This may be the easiest way out, although not necessarily the least expensive, as the die caster must, in the long run at least, be compensated for the extra operations he must perform and for such extra tooling as may be required.

In fastening die castings to sheet-metal parts, several alternative types of fastenings deserve consideration. They include: tapped holes in die-cast bosses; cored or drilled holes in die-cast bosses for self-threading studs or screws; studs die-cast in place; riveting with separate tubular or solid rivets, or fastening with through screws or bolts; riveting or spinning over of projections forming a part of the casting; use of so-called "speed nuts" on integral projections of the casting provided for the purpose; and use of hooked projections on the casting with a spring clip or other fastening for engaging one or more hooks; and various combinations of these types.

Naturally, the least expensive fastening which is satisfactory should be chosen and this consideration suggests checking against the foregoing list when there is a question as to this or some other fastening. In this connection, the following observations should be borne in mind:

With proper tools, correctly ground, tapping is very simply and rapidly done. But self-tapping screws and studs can be used with satisfaction, some of them being drive screws and others having standard threads, not necessarily hardened. They can be applied with a screw or stud driver, sometimes in a drill press or its equivalent. Use of studs die cast in place, although often an excellent construction, has the disadvantage of lengthening the casting cycle by a time equal to that for placing the studs in the hot die.

Although the use of separate tubular rivets (as in assembling the Oldsmobile-6 grille, for example) is entirely feasible and may be the best practice, as may be the alternative use of solid or split rivets in certain cases, these methods involve supplying and applying the extra rivets. It is often feasible and sometimes represents a saving, besides avoiding the use of exposed rivet heads, to cast the rivets as a part of the die casting, making the ends hollow or solid, as preferred. Thereafter, on assembly, the integral rivets may be struck or spun over (the metal being quite malleable) to afford a quick and secure fastening without any separate supply or handling of rivets. Often, the projecting integral rivet on a die casting is formed by merely drilling a hole in the die. Projections for speed nuts are formed similarly. Naturally, the location of the projecting parts must be such that the casting will clear the die. It is often a simple matter to form a small lip on a boss or other projecting part of a die casting – perhaps a part required for some other purpose – and afterward spin over this lip to assemble the casting to any mating part.

The radio grille on Buick cars, shown in Fig. 8, is an example of die casting which is fastened to another part – the instrument panel in this instance – by integral projections over which speed nuts are pressed. An example of the use of integral rivets is found in the small moldings on Ford glove-compartment doors. Such parts would be marred, of course, by through fastenings, but the integral ones are quite simple and effective.

Use of hooked projections is well illustrated in the headlamp molding employed on some Packard models. This curved molding is provided with five flat integral projections arranged to slip through corresponding holes pierced in the shell of the lamp. Each projection is undercut or hooked, an operation quickly performed by broaching, as it is not feasible to cast the undercut. The hook passes under the sheet metal at the end of each slot or hole and, since its face is sloped, it forces the die casting close against the lamp shell. There it is locked by a spring clip and is thus secure against loosening or rattles. If desired, one or more of the bosses could be arranged for a screw fastening, in which case a spring clip would not be required. This method of fastening might be adapted to almost any molding curved or straight instead of using a series of separate studs, and would save considerable time in assembly, especially where some screws, if used throughout the length, would come at inaccessible points. Fastenings of this type require that the casting be moved longitudinally a small amount when put in place and, in general, would need elongated holes, a part of which would not be filled by the projecting lug after insertion, but lugs would naturally be placed so that the hole is completely covered by the casting when the latter is forced home. Although a broaching or some equivalent operation is required for under-cutting the lugs, this operation should not be more expensive than drilling and tapping, and may even cost less as it can be done in a punch press. There are thus many reasons for anticipating wide use of fastenings of this type on certain classes of die-cast parts.

The fact that both Ford and Chevrolet, as well as many cars in higher price classes, employ die-cast tail-lamp brackets is, perhaps, sufficient indication of their economy and utility. In some cases, as in Buick models, the lamp housing itself is also die-cast and there is provision for supporting the license plate.

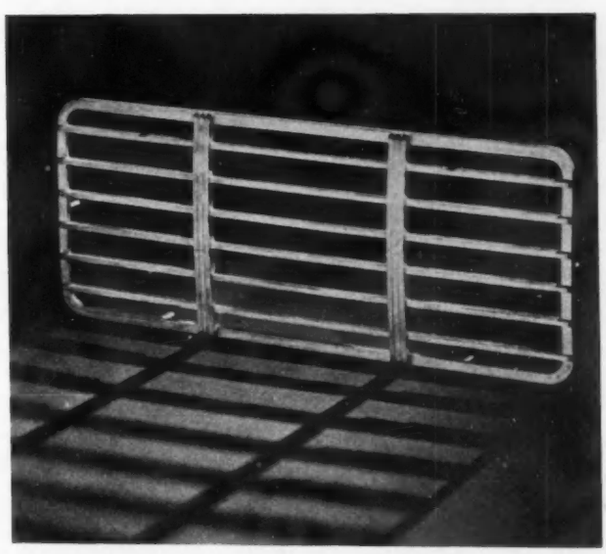

Fig. 8 – Buick Radio Grille Provided with Six Projections on the Back Face – In Assembly the Grille Is Held to the Instrument Panel by "Speed Nuts" Slipped Over these Projections

The fact that the die casting lends itself to production in complex forms cored out for light weight and in pleasing contours, often of streamline section is, of course, much in its favor. Fastening means are provided readily and can be concealed, and shapes which it would be difficult or impossible to stamp can be used. Much the same may be said of headlamp brackets, which are die-cast on several cars, and it is considered likely that headlamp bodies may yet be die-cast either separately or integral with their supporting brackets. Availability of new finishes suitable for die castings – finishes that are highly enduring and possess excellent adherence and which can be had in colors matching the body – have tended to promote the use of die castings requiring an organic finish, such as the brackets in question.

Frames for die-cast fender lamps are produced in numerous patterns and with attractive decorative effects. Recesses for lenses and other glass parts are cored out readily and very little if any machining is needed. Either plated or organic finishes are applied readily with little or no preparation of the surface.

Castings for Body Interiors

Aside from steering-wheel parts already dealt with, there are numerous other die-cast parts for body interiors which are deserving of mention. Brackets for attaching steering columns to the body are nearly all die-cast and constitute a good example of an important structural use. Cadillac is the only car which has yet employed an instrument panel which is completely die-cast, including the glove-compartment door, but this application is now in its second year and is a good example of a casting unusually long and thin in section as shown in Fig. 9. It is cast, of course, with the openings for the glove-compartment and instruments and is provided with the required fastening lugs. Many other cars have die-cast instrument frames and decorative parts for instrument panels. These include the radio grilles on Olds and Buick models. Several cars in the Chrysler lines have die-cast instrument-panel frames in which the slides for choke, throttle, and switch parts are guided and held flush with the panel.

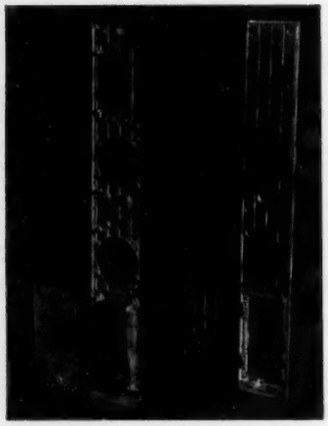

Fig. 9 – Instrument Panel and Glove-Compartment Door as Die-Cast in Remarkably Thin Sections for Use on Cadillac Cars – The Integral Fastening Bosses Are an Asset and Any Shape of Opening for Instruments Is Provided Readily

In applications such as these, in which the die casting often takes the place of stamped parts, it is usually the better appearance, the closer dimensional limits held, and sometimes the lower die cost which dictate the use of die castings. Much the same may be said of the housings and fittings for interior car heaters. Although some of these parts are provided with stamped shells, the latter are nearly always inferior in appearance to the die-cast shell and lack, of course, the convenience of integral bosses for mounting purposes. In addition, the die casting gives, as with most parts, a greater freedom in design, as it is possible to dispose the metal in ways which would not be feasible, if possible at all, in stamped units. Heater fittings for attachment of tubing to defroster outlets are also die-cast to good advantage as is the housing for the fan.

When it comes to hardware, exterior as well as interior, die-cast parts are practically without competition and give universal satisfaction. Again the matter of appearance and freedom in design, as well as the remarkably smooth finish which is attained today, are all in favor of the die casting. Recently it has been possible to produce acceptable hardware parts cast so smooth that even buffing the surface before plating is dispensed with. Now that a satisfactory bright-nickel finish has been developed, it is also possible today, nearly if not entirely, to do away with polishing, buffing, and coloring operations for most grades of hardware. Ford is continuing the use of some Macoid finish on interior hardware, and this process also does away with polishing and buffing.

Much experimental work has been done, both here and abroad, in the coating of hardware parts produced by die casting with cellulose acetate applied by the injection-molding process. This method has attained a commercial application in the electric refrigeration field but is not yet used for automobile hardware to the author's knowledge. It permits of extremely attractive finishes, which give promise of good enduring qualities, but it remains to be seen whether it can compete with plating in cost.

Die-cast hardware is constantly undergoing style changes which are rendered less expensive than they would be otherwise because of moderate die cost. This year there has been a marked change toward heavier hardware, dictated entirely by style considerations. An effort is being made, which is commendable, to provide hardware which does not catch in clothing and which, in interiors, projects as little as possible above the surface on which it is mounted. This change has been made with a view to reducing the chance of serious injuries when an accident or emergency turn or stop may throw passengers against projecting parts. Some hardware, especially outside door handles, could be made lighter in weight by coring or other design changes, but stylists are generally more interested in appearance than in weight saving.

Not many years ago, the bells and other parts of horns were produced largely from stamped and drawn parts. Today, both bells and bodies, some with quite intricate spiral passages, are

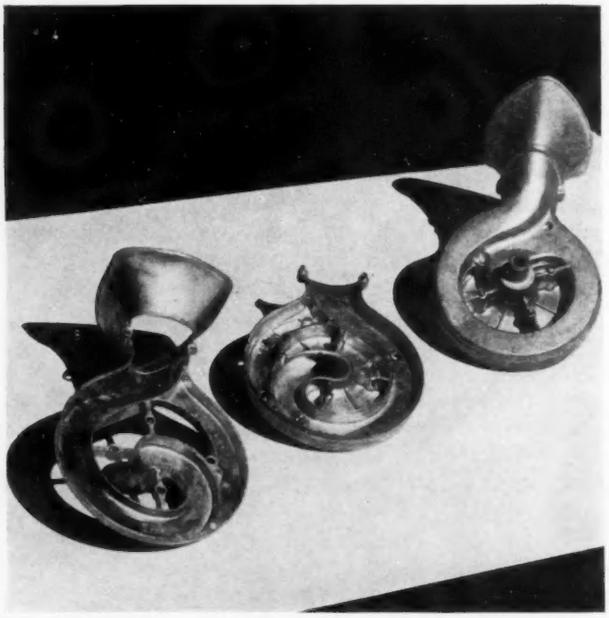

Fig. 10 – Die-Cast Horn Parts One of Which Carries Integral Projections Which Pass through Mating Holes in the Other Half and Are Afterward Spun Over to Fasten the Parts Together

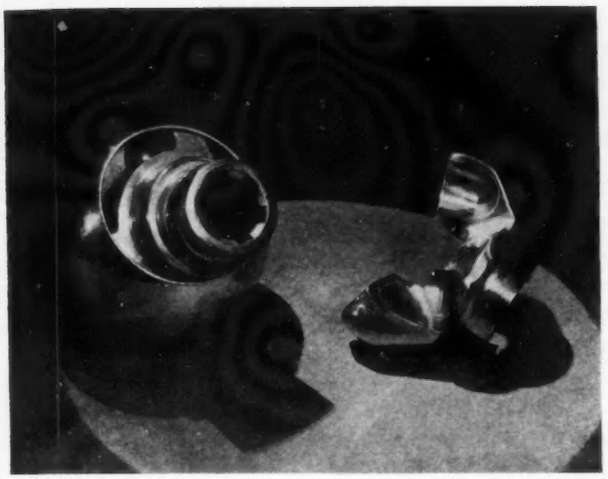

Fig. 11 – Bendix-Weiss Universal Joint in Which a Die-Cast Matrix of Zinc Alloy Accurately Positions the Hardened-Steel Ball Races in Reference to the Forged-Steel Yoke

die-cast as shown in Fig. 10. One important reason for this change is that the die casting is not only less expensive but, in the case of some designs, is the only feasible method of production. Another reason is that tone qualities are secured with the zinc alloys superior to the stamped sheet formerly used. Although the die-cast horn probably weighs more than one built up from stampings, the former is produced with quite thin sections, and the saving in production cost is not to be overlooked.

With a stamped product, several parts are required and considerable soldering is needed in assembly. Drawing operations in stampings require expensive dies in addition to blanking dies, whereas the single die for die castings costs less initially and is maintained more easily. In the modern application of horns, hidden under hoods or fenders, finish is not so important as it was when the horns were displayed prominently in front of radiators. Finish was then important and was usually superior with the die-cast horn. In this connection, it should be remembered that a drawn part not infrequently shows a rippled surface which, when plated and polished, shows the irregularities prominently. Some such parts also tend to go out of shape when buffed. These disadvantages are not encountered with the die casting.

Reference to Fig. 10 shows the horn castings to be split in the axis of the bell and the spiral leading into it. The mating faces have to make a tight fit and are held together by several screws or bolts or, in some designs, by rivets which are made integral with one casting and afterward spun over when the parts are assembled. In such applications, or any others in which the zinc alloy is put under compression by tightening the screws, it is well to remember that the zinc is subject to a certain small creep or cold flow. For this reason, the thickness of bosses should be minimized and it is wise to use a spring washer, as there is some tendency for screws to loosen. This loosening has caused trouble where precautions against it have not been taken but, with proper design, it is not serious as otherwise die castings could not be used in many automotive applications, such as the horns in question and fuel pumps, of which many millions are in use.

Chassis Applications

A large majority of the applications of die castings to chassis parts are so well known and so much taken for granted as not to require extended comment here. Such parts as carburetor bodies, fuel pumps and filters, lamp brackets, accelerator parts, oil-seal rings for crankshaft bearings, distributor brackets, steering column brackets, shifter-lever trunnions and balls, running-board fittings and steering-wheel hubs either have been mentioned before or are so commonplace that further mention is not necessary.

Somewhat less common, but of growing importance, are such parts as end plates for vacuum-brake cylinders and several similar or supplementary castings in vacuum cylinders and housing parts on vacuum-operated gear shifters and clutch actuating elements. The reasons for using die castings in such parts are much the same as for other chassis parts, and so need not be detailed here.

An unique application of die castings developed recently to solve a difficult problem, however, is that of the Bendix-Weiss universal joint illustrated in Fig. 11. This joint, instead of using the conventional trunnion, carries a steel ball between races supported by the yokes of the driving and driven portions. The races are hardened steel and the yokes are steel forgings with splined hubs. To make the races integral with the hubs involved production problems such as to make it difficult to compete with joints of more conventional design. It is relatively simple, however, to make the hardened races separately. The problem was to secure them to the yoke in precisely correct position at moderate cost. An excellent solution of the problem was found in the use of zinc alloy die-cast in such a way as to form a matrix joining the respective parts. In other words, the yoke and the two races are held in correct relative position in a casting die, and the space between and around them is filled with the zinc alloy by the die-casting process. As the zinc alloy is in compression, in which it has a strength of at least 60,000 lb. per sq. in. (some zinc alloys run as high as 93,000 lb. per sq. in.), it is understood to meet all requirements, even under heavy overloads, shock loads, and low temperatures.

Although this is an unusual application, it may well suggest others in which zinc alloys may be used in combination with other metals employed as inserts to provide properties which the zinc alloy itself cannot meet. As the casting temperature of zinc alloys is under 850 deg. fahr. and they cool very rapidly in the die, the temperature of inserts is not raised enough to soften a hardened steel part or, in fact, to affect adversely such materials as phenolic plastics, wood, or even compressed paper, all of which are sometimes used as inserts, although not, as far as the writer can recall, in automotive parts. It is a common thing to insert a bronze bushing or an insert of steel in die castings and there is little doubt that, by the intelligent use of inserts, many die castings might be improved or made lighter. There is still plenty of opportunity for the designer to use his ingenuity to advantage along this line.

Plated Coatings and Organic Finishes

No paper on the general subject of die castings in the automotive industry can be considered complete without reference to the subject of finishing. At the present time most finishing on automotive parts is of the plated type. In recent years practically the entire automotive industry has standardized on certain definite specifications for plated coatings on zinc-alloy die castings. Assurance has been given by the leaders in the industry that electro deposits satisfying these specifications are giving good service.

It is not necessary to go into detail as to the individual specifications of the various companies. In general, they all require heavy coatings, the minimum varying from 0.0008 to 0.001 in. This coating may be composed of a series of copper, nickel and chromium, or of nickel and chromium, the option being allowed in most cases. Where the sequence is copper-nickel-chromium, the minimum copper thickness is specified variously as 0.0002 and 0.0003 in. The nickel deposit is variously specified as a minimum of 0.0003 and 0.0005 in. The chromium in most cases is required to be a minimum of 0.00001

in., although one large producer requires 0.000025 in. It cannot be emphasized too strongly that the satisfaction now being obtained with plated die castings in the automotive industry depends upon the conscientious fulfillment of these specifications.

In addition to the thickness of coating specifications, most automotive concerns also specify some type of quality test such as salt spray or contact with hydrogen sulphide. Tests of this type are intended to detect major deficiencies in plating quality.

One cannot leave the subject of plating specifications without calling specific attention to the fact that they are based on definite experimental and service evidence compiled by a number of capable workers in the industry, and that they are not figments of imagination, but definite operating limits based on experience.

With the exception of the use of color as a contrast for chromium plate in ornamental parts, not many automotive die castings are subjected to organic finishing. The details of the procedures used in specific instances are not available. It is definitely known, however, that at least one finishing system calls for the use of a suitable primer over which either an air-drying or baking finish may be applied. Other systems on which experimental evidence is now available permit the application directly of baking enamels on the zinc-alloy die castings. In this system the die castings are pretreated in a phosphate-type solution. This pretreatment makes it possible to select the desired finish on the basis of color matching, coating properties, and so on, without consideration as to its applicability to zinc.

Notes and Suggestions on Design

In the foregoing comments on current applications there are indicated or implied numerous "do's and don'ts" which the designer should observe for optimum results with die castings. There are, in addition, many general observations, some of which may appear trite but which, nevertheless, are so frequently overlooked that repetition is fully justified. Some of these and certain other recommendations are given in following paragraphs, a review of which may well repay any designer who is about to lay down a new design for a die casting. The need for considering such items as these is apparent if the views of one engineer, whose duties require that he criticize and strive to correct the faults in suggested designs for die castings within his own company, are to be taken at face value. He says in effect that two out of three designs as submitted to him are regarded as unsatisfactory, usually because some more or less commonplace requirement for best results has been overlooked or brushed aside. Whether over-stated or not, it is at least apparent that more careful consideration of items such as are listed in the following paragraphs would avoid much "grief" for all concerned.

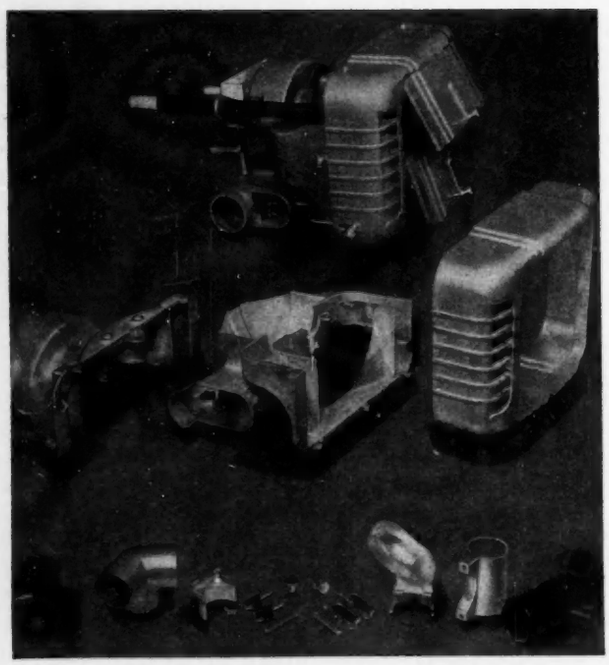

Fig. 12 – Well-Designed Car Heater in Which the Main Housing and Several Auxiliary Parts Are Made in Die-Cast Form, Yielding Advantages in Appearance and in Convenience in Mounting and Assembly

As in other types of castings, both inner and outer corners and edges should have fillets; the larger the better. Although sharp corners can be cast, they often if not always tend to disrupt the smooth flow of metal and are likely to result in defects. These defects may include increased porosity and surface blemishes productive of much trouble in finishing and resulting in much more numerous rejects, often after considerable time has been spent in polishing and plating. When, for esthetic reasons, the stylists insist on sharp edges, even a very small fillet, hardly visible except on close inspection, may help materially. Omission of fillets often results also in breakage otherwise avoided, and also make polishing and buffing more difficult. Plating also is affected adversely.

Where buffing is required on any surface of the casting, as it usually is before plating, the designer should try to visualize the position that the casting must assume relative to the wheel and design the surfaces so that they are reached readily by the buffing wheel. Depressions, when required should, so far as possible, lie parallel to the direction of motion of the casting in reference to the wheel, should be as shallow as conditions permit and should have sweeping fillets where the depression merges with other surfaces. Ribs or beading likewise should be parallel with the normal motion of the casting against the polishing wheel and have liberal fillets where they join other surfaces. Convex surfaces are usually much easier to buff than concave ones. A surface of rotation is usually easy to polish if not interrupted by projecting parts. Thus, a circular steering-wheel hub is polished readily on a rotating fixture but, if its surface were interrupted by projecting spokes, the buffing obviously would be more difficult.

Large plain flat surfaces should be avoided if exposed to view in service. They are likely to be wavy or to have blemishes which are not easy to remove. Even slightly crowned surfaces may tend to avoid such defects, especially as they give concentrated highlights when plated and polished. Stepped or relieved surfaces, well-rounded beads, or bands of beads, often improve appearance greatly and tend to mask defects, of no consequence structurally, but tending to stand out prominently on a flat surface. The casing of the car heater illustrated in Fig. 12 is an example of good design including well-rounded corners and the use of steps and beading in such a way as to enhance appearance without departure from simple and attractive forms. Incidentally, when constructing a die for a part of this kind, it is often much better to work from a dummy or model than from a drawing in which the curves have to be defined geometrically. Where fastenings or other locating parts are to be held within close limits, of course, these dimensions are better located from a drawing.

Sudden changes in section thickness should be avoided so far as possible but, where not avoidable, the largest possible fillet should be used or the transition should be made as gradually as possible. Weight often can be saved and stiffness improved by judicious use of ribs and bosses, or by reinforcement at fastening points.

Perhaps there is no general rule so frequently stated or so

(*Continued on page* 314)

Automatic Transmissions

By Austin M. Wolf
Automotive Consultant

Part I - The Cavalcade of Transmission
Part II - Bus-Operating Experience with the "Mono-Drive"

STARTING with the Sturtevant in 1904, transmission developments are reviewed to date. Requirements for successful and satisfactory automatic transmissions are discussed and interpreted in the light of present and future development.

Six tables that classify transmissions comprise the nucleus of this part. These tables group the methods of varying the speed; the means of control; the gear engagement or connection; the shifting methods; clutches; and clutch control.

THE story of over two million miles of operating experience using the "Mono-Drive" type of automatic transmission on 101 passenger buses in Chicago, is told in the second part of the paper. The mechanical operation of this automatic transmission is explained, and specifications and data are given on the buses in which they are used. A review of the maintenance record of the transmissions concludes the paper.

IN the development of the motor vehicle, intensive concentration on engine design, construction, and materials over the major portion of its evolution has resulted in the advanced status of today's powerplant. There can be no doubt that at the present moment the chief unit of interest on the drawing-board, in the laboratory, on the road, and on the proving-ground is an improved form of transmission over what we have been accustomed to, and which also will adapt itself better to future vehicle designs. The original "clash gear" was considered an unworkable monstrosity which, however, survived practically unchanged other than the conversion from the progressive to the selective type of mechanism until the recent addition of constant-mesh gear trains, synchronizers of various types, overrunning clutches, automatic clutch actuation, and power shifting. The last with its decisive action also has overcome some of the shortcomings of the manual shift.

[This paper was presented at the Metropolitan Section Meeting of the Society, New York, Feb. 8, 1937.]

General Types

Automatic transmissions can be classified generally as those of the stepped type (fixed ratio) and those having an infinite number of ratios, with positive or positive and negative directions of rotation. There are many who believe that, should the infinite-ratio transmission become available, the requirements for an automatic control thereof would be superfluous. In view of the ability to correlate the engine and the rear wheels by means of any desired ratio (within usable limits), the "feel" obtained by small increments of speed might enable the operator to select with better judgment than in the case of widely stepped ratios. There are necessarily certain inherent characteristics of each type of transmission mechanism which also more or less will determine whether or not pure automatic control would be most advantageous. In some systems its incorporation adds little or no complication. It is certainly better to get a more nearly correct ratio at all times than a prevailing poor one. We must hold the proper viewpoint and attempt to detach some of our thinking from what might be termed an engineer's driving technique as compared to that of Mr. John Q. Public. We have only to look back on the automatic spark advance and the automatic choke to decide that, for all-around performance, they achieve far better results than could be obtained from an operator who knows little of "what it is all about." Furthermore, man is instinctively lazy and ease of operation is always a desirable and quickly accepted feature.

There has been considerable discussion regarding the ability of a fully automatic transmission control to "think." There can be no question regarding the inability of one given mode of operation satisfying each and every operator. Until more complete and comprehensive information is available based on actual experience with all types of individuals, it would appear that the addition of a manual control that can assert itself over an automatic transmission would be desirable. Furthermore many operators wish full control rather than to relegate it to a robot. Probably torque controls have been experimented with more generally than any other type, in the case of stepped transmissions. Their reliability has been questioned as they have a tendency to make a shift when the operator does not desire it or not to shift when he so wants it. In many cases they have been found to get out of adjustment very easily. A more practical and satisfactory method seems to be one based on car speed. The centrifugal governor, however, must be so

Table 1 – Methods of Speed Variation

Variable-Speed Transmission

(I) Mechanical
- (A) Frictional Contact
  - (a) Belt (pioneer cars)
  - (b) Friction discs at right angles (Orient, Carter-car)
  - (c) Grooved wheels (Duryea)
  - (d) Metallic rollers (Hayes, Transitorq, Graham)
- (B) Gearing or Chains
  - (a) Sliding-gear transmission
    - (1) Progressive
    - (2) Selective
  - (b) Chain drive between shafts
  - (c) Planetary
- (C) Links or oscillating arms with ratchet devices
  - (a) Eccentric, link, pawl and ratchet
  - (b) Variable-throw, roller clutch (DeLavaud)
- (D) Inertia mechanisms (Constantinesco)
- (E) Centrifugal force (Spontan, Hobbs)

(II) Electric
- (A) Straight conversion (Yellow Coach, Mack)
- (B) "Feed-back" circuit (Entz)
- (C) Storage battery "floating" (Woods Electric)

(III) Hydraulic
- (A) Radial pump and motor
  - (a) Variable-throw crankpin (Manly)
  - (b) Variable path for outer rollers (Hele-Shaw)
- (B) Multiple pump and motor (Pitter)
- (C) Pump acting as clutch with "feed-back" circuit
- (D) Hydraulic torque converter (Lysholm-Smith)

(IV) Pneumatic (Exhibit at old Boston Show)

(V) Electro-mechanical (Electro-Gear)

(VI) Hydro-mechanical (Bendix)

Variable-Crank-Throw Engine

designed into the assembly, in view of space and weight limitations, that it must be able to cope with all the demands that will be placed upon it, unless a servo mechanism is added. Should the ordinary governor be sensitive to low speeds, it is liable to be too heavy and impractical for high speed. The use of more than one governor seems to alleviate this condition.

Definition

A transmission, in its broad term, includes the entire mechanism between the engine crankshaft and the rear wheels – from the source of power to the point of propulsion at the road contact. Although the gear-box often has been termed a transmission, the various types and methods of transmitting power warrant this broad interpretation when necessary. Clutch developments, overdrives, auxiliary gear boxes, and two-speed axles indicate but a few of the variations that are likely to come into this discussion as well as unconventional methods of obtaining a variation in speed with a corresponding inverse torque multiplication.

Positive Type

A positive-displacement type of transmission mechanism requires that, when a certain part of the mechanism is actuated, all other parts will be displaced definitely regardless of speed, as in the case of a gear train, transmission of power by chains, levers or links, a positive clutch, or the use of a positive-displacement type of hydraulic pump. There is also a "build-up" type in which a time lag or speed element enters, as in the case of a centrifugal clutch, a Fottinger Coupling, a generator-motor combination, a compressor-air-motor combination or a transmission system based on inertia or centrifugal force. Positive displacement ensues with some, whereas a small constant slippage is inherent in others.

Classification

In order to picture the several types of developments of design, it will be interesting to refer to Table 1 which groups the various methods of varying the speed between a driving and driven member. It will be noted that the variable-crank-throw engine is grouped separately as this method cannot be classified as a transmission. It would seem from this analysis that all of the conceivable general methods of possible power transmission have been tried. There is no doubt that various combinations can be tied together and that new versions of old attempts will reappear.

In Table 2 we classify numerous means of controlling a transmission, including forces to which the control mechanism can be made sensitive. There are bound to be many other means in this field which will turn up as the developments covered by Table 1 extend further. These transmission controls can be incorporated in either the stepped type or the infinitely variable transmission.

Whether gear engagement or a driving connection is to be established, the various means shown in Table 3 have been tried. To cause such actuation, the methods in Table 4 have been used.

Having resolved the various classifications and systems into their elements, I am in hopes that this picture possibly may

Table 2 – Transmission Control

(I) Manual
- (A) Direct
- (B) Preselective (Commer Truck)

(II) Outside Control, sensitive to
- (A) Car speed-centrifugal force
  - (a) Servo governor (Bendix, Vaco)
  - (b) Direct actuation (Reo)
- (B) Engine speed-centrifugal force (Sturtevant)
- (C) Torque reaction
- (D) Intake-manifold vacuum
- (E) Throttle position
- (F) "Gear" or speed position for next step
- (G) Combination of
  - (a) (A) and (C) (Sneed)
  - (b) (B) and (E) (Fleischel)
  - (c) (A) and (F) (Vaco)
- (H) Reversal of torque (Mono-Drive, overdrive)

Table 3 – Transmission Gear Engagement or Connection

(I) Sliding gear

(II) Constant-mesh gear
- (A) Sliding positive clutch member
- (B) Friction pickup
  - (a) Cone clutch (Winton, synchromesh)
  - (b) Multiple-disc clutch (Matheson, Welsh)
  - (c) Internal-expanding clutch (Haynes, Mono-Drive)
  - (d) Electro-magnetic (Cotal)
  - (e) Hydraulic pressure (German transmission)
  - (f) Pneumatic
    - (1) Vacuum
    - (2) Compressed air (Northern, Fiat)
- (C) Overrunning clutch
  - (a) Pawl and ratchet
  - (b) Roller clutch
  - (c) Coil spring

Table 4—Transmission Shifting Methods

(I) Manual

(II) Pneumatic
- (A) Vacuum
  - (a) Solenoid control (Bendix)
  - (b) Direct (Maybach)
- (B) Compressed air (Westinghouse)

(III) Electric
- (A) Solenoid (Cutler Hammer)

(IV) Hydraulic pressure
- (A) Engine lubricating system
- (B) Engine-driven hydraulic pump

inspire some of you toward accomplishing the next Utopian step. Mechanical solutions owe much to the recent developments of a successful centrifugal clutch, torque balance phases due to a differential hook-up, and the replacement of the old pawl and ratchet by the overrunning clutch of the roller type.

Table 5—Clutches

(I) Positive
- (A) Dog type
- (B) Gear type
- (C) Sliding key (Haynes, Apperson, Chandler "Traffic")

(II) Friction
- (A) Wedge type
  - (a) Cone
  - (b) Hele Shaw
- (B) Flat-plate type (multiple disc, wet or dry; single or double plate)
- (C) Cylindrical-surface type (internal expanding, external contracting)
- (D) Coil spring (Mercedes, L.G.S.)

(III) Hydraulic
- (A) Turbo-type coupling (Radcliffe, Fottinger)
- (B) Clutch (pump with valved bypass)

(IV) Electric

(V) Electro-magnetic (Cotal)

(VI) Pneumatic

(VII) Combinations of above

Table 6—Clutch Control

(I) Manual (customary engagement by spring pressure)

(II) Centrifugal
- (A) Full torque capacity at relatively high speed (Power Flo)
- (B) Full torque capacity at low speed (Mono-Drive)

(III) Pneumatic
- (A) Vacuum
  - (a) Servo
  - (b) Automatic with accelerator hook-up
- (B) Compressed air (Westinghouse)

(IV) Hydraulic, servo
- (A) Pressure from engine lubricating pump
- (B) Separate engine-driven pump (English and German)

(V) Combinations of (I) and (II)

Intermittent or Constant Torque

The stepped transmission with manual control, during speed changes, gives an intermittent application of torque to the point of road contact and is a considerable handicap in the case of heavy vehicles in which the rolling resistance is high, such as track-laying vehicles over soft terrain. A different condition exists in the ordinary motor car which rolls readily, and a shift at the reversal of engine and coasting torque (zero torque) provides the ideal point for a change in stepped ratios to avoid jar. There are transmissions with power shifting in which speed changes occur so quickly and without much jar or hesitation that the cessation of power application is very brief. More work and study are necessary along these lines before the final answer is found as to the superiority of a quick shift over a slow one. Under certain particular conditions there are strong and undeniable advantages to the continuous-torque type of transmission.

Clutches

In view of the wide use of a clutch mechanism of some type, a similar analysis of types and control is of interest and is given in Tables 5 and 6.

It seems essential in a centrifugal clutch that it be able to take care of full engine torque even at the lower engine speeds. This function cannot be done if the force exerted by the weights be the sole means of applying pressure, unless a very heavy construction be used.

Review

A retrospect covering only a few of the numerous transmission developments of interest follows. Some of the high spots will be recognized readily:

Sturtevant	1904	Centrifugal control, engine speed.
Manly Drive	1908	Hydraulic transmission.
Hele Shaw	1912	Hydraulic transmission.
Owen-Magnetic	1914	Electric transmission, "feed-back."
Radcliffe Drive	1918	Similar to Fottinger Coupling.
Daimler Fluid Flywheel	1926	Fottinger coupling (original pre-War).
Constantinesco	1926	Inertia transmission.
DeLavaud	1928	Variable-throw transmission.
Spontan	1930	Centrifugal-force transmission.
Banker	1933	Car speed and coasting torque.
Hayes	1933	Friction roller on toroidal surfaces.
Transitorq	1935	Friction roller on toroidal surfaces.
Reo	1933	Centrifugal control, car speed.
Electro-Gear	1934	Electro-mechanical.
Lysholm-Smith	1934	Hydraulic torque converter.
Bendix Electric Hand	1935	Electro-vacuum shift control.
Sneed	1935	Torque and speed.
Hobbs	1935	Centrifugal-force transmission.
Bendix	1936	Hydro-mechanical transmission.
Weiss	1936	Hydro-mechanical transmission.
General-Electric	1936	"Public Service" electric transmission.*
Studebaker	1936	Overdrive, coasting torque.

*Latest type with Diesel powerplant.

One of the latest designs covering the variable-throw-crank engine is the English design by Thomas, disclosed in 1935 as a six-cylinder engine with a bore and stroke of 70 mm. and 101 mm.

It is my sincere hope that this review will create a still greater interest in what probably is a most enticing field of development today. Metallurgical improvements and manu-

facturing facilities that were undreamed of a few years ago, coupled with a little ingenuity, will no doubt tear down what barriers do remain in the way of attaining the perfect transmission. When this job is done, the engine builder will have a lot of new problems on his hands. That, however, is a different story and will be just another link in the chain of automotive progress. The economy consciousness (small engine) that has at last been awakened focuses a lot of future development on the solution of this transmission problem. "Give me a lever long enough and I will lift the world," said Archimedes. Our slogan is: "Give me a lever long and short enough, with a readily variable fulcrum point, and I will lift the world to a new plane of car performance."

Part II

TWO million miles of operating should impress anyone with the fact that the operator has had plenty of time and experience to become familiar with his equipment. This story is of all the more interest since the buses are equipped with automatically controlled transmissions.

The Chicago Motor Coach Co. now has 101 new coaches of the double-deck type equipped with the Mono-Drive. Between July 16, 1936, and Jan. 16, 1937, the total mileage reached 2,006,679 as shown in Table 1. The initial production coach, No. 146, has covered 34,343 miles in six months. Its route measures 27.34 miles for a round trip and it is estimated that it makes 6000 stops and starts in one day's run of 9½ trips (259.73 miles), consuming 18 hr. (total average speed 14.4 m.p.h.). Although none of the lines is equipped 100 per cent with this new equipment, increases of up to 40 per cent over last year in passenger traffic has resulted on the North Division.

Service Observations

There has been an increase in safety as indicated by the reduced number of accidents, decreased driver fatigue by the elimination of thousands of gear shifts and declutchings during a day's work; yet the schedules have been maintained with greater ease in spite of the greater number of starts and stops required by the increased number of passengers. The smoother operation has been noticed distinctly by the passengers who avoid the old type of equipment. It is so pronounced that an engine that is not tuned up properly is noticed immediately and turned over to the maintenance department. The "feel" of the motor is not necessary in shifting, and this advantage is a distinct contribution in a rear-engined bus where microphone "eaves-dropping" on the engine has been resorted to by one operator.

During this period not a single universal joint, driveshaft, or axle shaft has been broken due to shock loads. This record is of particular interest in a rear-engined installation where the close coupling of the units prevents the desirable resiliency of the previous long transmission line and where the engine is ordinarily "reved up" to much higher speeds than in a bus with a forward engine location.

The Model 720 Coach

The Model 720 G.M. Coach seats 72 passengers, 31 on the lower deck and 41 on the upper. The body construction is mostly of aluminum. The powerplant consists of a 707 cu. in. engine developing 173 hp. at 2100 r.p.m. Maximum torque is 550 ft-lb. Two transverse springs are used fore and aft of the front-axle center section and are 57 in. long and 3 in. wide. The rear springs are semi-elliptic, 64 in. by 4 in. The rear-axle ratio is 3.92:1, and the overall ratio through the angle drive is 6.59:1. A 100-gal. gasoline tank is mounted ahead of the engine compartment, entirely outside of the passenger compartment. An Electro-Acoustic Product Co.'s announcing system is used with a microphone located close to the driver and a loud speaker at the rear of the lower deck and one at the front of the upper deck. Other data are given in Table 2. An exterior view of the bus is shown in Fig. 1, whereas Figs. 2, 3, and 4 give interior views. These coaches occupy 3.7 sq. ft. per seated passenger as against 7.4 sq. ft. per seated passenger in the latest type of surface cars.

The Mono-Drive Main Clutch

It may be recalled from Part I of this paper that the Mono-Drive consists of a planetary type of transmission mechanism with three speeds forward and one reverse and with automatic control.

The automatic clutch shown in Figs. 5 and 6 consists of two internal shoes which are expanded by centrifugal force through governor weights and their connecting linkage and two pressure springs. The clutch drum which the shoes contact is part of the transmission assembly. Two centrifugal weights are used and they are connected through a primary toggle to an equalizer, from which point a link extends to the secondary toggle mechanism that expands the shoes at each end. The clutch starts to engage at about 400 r.p.m. of the engine, and

Table 1 – Model 720 Double-Deck Coaches – Accumulated Mileage to Jan. 16, 1937

North Division		South Division		West Division	
Coach	Mileage	Coach	Mileage	Coach	Mileage
104	46,796	*146	34,343	175	23,693
105	20,154	147	31,607	176	23,457
106	19,169	148	29,645	177	22,894
107	18,018	149	29,144	178	22,129
108	19,757	150	32,756	179	21,853
109	20,965	151	25,489	180	24,437
110	19,919	152	30,704	181	20,702
111	20,402	153	29,321	182	22,673
112	19,930	154	26,523	183	21,329
113	19,819	155	31,951	184	22,677
114	16,076	156	26,402	185	21,846
115	19,779	157	22,812	186	20,431
116	19,523	158	24,310	187	11,075
117	18,971	159	24,815	188	22,230
118	20,910	160	23,928	189	19,288
119	18,309	161	22,903	190	20,496
120	19,149	162	24,135	191	17,764
121	19,867	163	23,073	192	18,801
122	21,491	164	25,929	193	17,172
123	18,000	165	22,699	194	17,132
124	20,757	166	20,044	195	17,796
125	18,352	167	17,619	196	16,584
126	17,683	168	16,403	197	18,229
127	17,506	169	17,660	198	17,781
128	16,614	170	17,704	199	15,726
129	16,231	171	16,945	200	16,424
130	17,796	172	16,226	201	11,608
131	16,300	173	16,319	202	9,126
132	18,027	174	16,723	203	11,841
133	17,690			204	12,266
134	15,780				
135	14,631				
136	14,687				
137	12,809				
138	14,069				
139	14,645				
140	13,962				
141	12,672				
142	11,504				
143	12,514				
144	10,008				
145	7,846				
	749,087		698,132		559,460

Model 720 Fleet Mileage 2,006,679

*No. 146 (initial production coach) placed in service July 16, 1936.

Figs. 1 to 4 – Views of the Model 720 G.M. 72-Passenger Coach

Fig. 1 – Exterior
Fig. 2 – Lower Deck from the Front
Fig. 3 – Lower Deck from the Rear
Fig. 4 – Upper Deck from Front

the maximum effort of the weights is reached at 800 r.p.m., at which speed the clutch is engaged fully. Any additional pressure of the shoes on the drum is due to centrifugal force acting on the shoes themselves. The clutch is 16 in. in diameter and 3¼ in. wide.

Since there is a slip period between 400 and 800 r.p.m. when accelerating, excessive slipping is prevented by pushing the accelerator way down as far as it will go without hesitation. The clutch remains fully engaged until the engine speed again drops down to 400 r.p.m. after which it snaps out of engagement. This action is due to the fact that the pressure spring is locked in by the toggle which is nearly straight, preventing the reaction of the spring reaching the centrifugal weights which are provided with torsion springs surrounding their hubs for their return. The pressure springs are preloaded to approximately 1400 lb. each in the "off" position, and the anchor end is pinned to the shoe. The roller at the toggle end creeps on the inner face of the shoe and is able to compensate for lining wear. Torque to the shoe is transferred from the engine flywheel spider plate through a centrally located trunnion block. The toggles serve no function except to expand the shoes into the engaging position and to augment the force of the pressure springs with the force of the weights during this period. The weight travel is limited by a positive stop. The secondary toggle pin has a head extension on which is mounted a sleeve roller centered in a radial slot of the spider. The shoes are thus centralized, and a definite clearance relationship between the lining and the drum is maintained. The retraction springs help in this function.

The radius of the shoe is 0.015 in. less than the drum diameter in order to obtain easy initial engagement which is distributed over approximately a 90 deg. contact before the full spring pressure is exerted. Full contact is obtained by the distortion of the shoe whose rib contour is designed definitely for this purpose. Any semblance of chatter or grabbing is

Table 2 – Dimensions Model 720 Coach

Overall length (over bumpers)	33 ft. 0 in.
Overall length (body)	32 ft. 9 in.
Overall width (over tires)	95¾ in.
Overall width (body)	95 5/16 in.
Overall height (loaded)	12 ft. 8½ in.
Overall height (unloaded)	12 ft. 11½ in.
Floor or step height (unloaded)	16½ in.
Headroom (lower deck)	74 13/16 in.
Headroom (upper deck)	62⅜ in.
Wheelbase	217 in.
Track (front)	84 in.
Track (rear)	72 in.
Tire size (front)	10.50-22
Tire size (dual rear)	10.50-24
Turning radius (over corners)	42 ft.

eliminated entirely. The springs are compressed to approximately 1600 lb. each by the toggles and the quick release when the engine speed reaches 400 r.p.m., due to the withdrawal action of the weight retraction springs snapping the primary toggles from their near dead-center position.

The Transmission

The three-speed transmission provides ratios of 3.25:1, 1.56:1 and 1:1 in forward speeds and 3:1 in reverse. At the left end of Fig. 7 is shown the clutch drum which drives the first-speed shaft through an overrunning clutch adjacent to its hub. The second-speed shaft is tubular and surrounds the first-speed shaft. The gears at each end of these shafts mesh with three sets of planetary gear clusters. When the planetary housing is stationary, drive is conveyed from the wide-faced gear of the cluster units to the main driveshaft and thence by bevel gearing to the rear axle.

Between the planet housing and its front bearing, a one-way roller clutch known as the reaction clutch prevents the housing from turning backwards around the sun gears. First speed is effected by stepping quickly on the accelerator in order to bring the automatic clutch into engagement with the main clutch drum. Power then flows through the overrunning clutch, its inner race which is splined to the first speed shaft, and thence through the planetary gear.

Due to the constant-mesh relationship of the first- and second-speed sun gears, it is apparent that, while the first-speed reduction is effective, the second-speed sun gear is being driven by the cluster units. At the left end of the second-speed shaft is a centrifugal governor which, as the vehicle speed increases, will overcome the force of two compression springs and attempt to move the second speed clutch member to the left through the intermediary of a coil spring. The mating clutch member is secured to the main clutch drum. The faces of the lugs on the driven clutch member are relieved at the trailing edge, and the drive lugs are relieved at the leading edge, causing the lugs to slip as the governor weights attempt to effect engagement. Since the clutch drum revolves faster than the driven clutch member, the former overrides the latter. When the accelerator is released to effect the gear change into second, the clutch members go into full engagement as the main drum decelerates to a point where the speeds of the driving and driven clutch members synchronize. Power is now conveyed from the main clutch drum directly through the second-speed clutch on the second speed tubular shaft and then through the second-speed gearing of the planetary system. While in second gear, the planet cluster units drive the first-speed sun gear and its shaft faster than the main clutch drum. The overrunning clutch therein permits this "over-run."

It will be noted that the planet housing has not been revolving during first and second speeds, being restrained by the reaction clutch member. Its outer race has the cam contour on its inner face against which the rollers abut and on the outside is provided with gear teeth, forming the male member of an internal-external gear clutch. The outer member in Fig. 7 is in the neutral position.

During forward drive the non-revolving internally toothed clutch member engages the reaction clutch cam-gear when it is shifted to the right. This clutch member is supported on a carrier plate, sandwiched between the front and rear sections of the transmission case. It also contains a large ball-bearing, supporting the front end of the planetary case. The clutch member is mounted on and guided by an externally toothed hub portion of the carrier plate and is able to engage the reaction clutch cam-gear by means of segments passing through slots in the carrier plate as shown in Fig. 8.

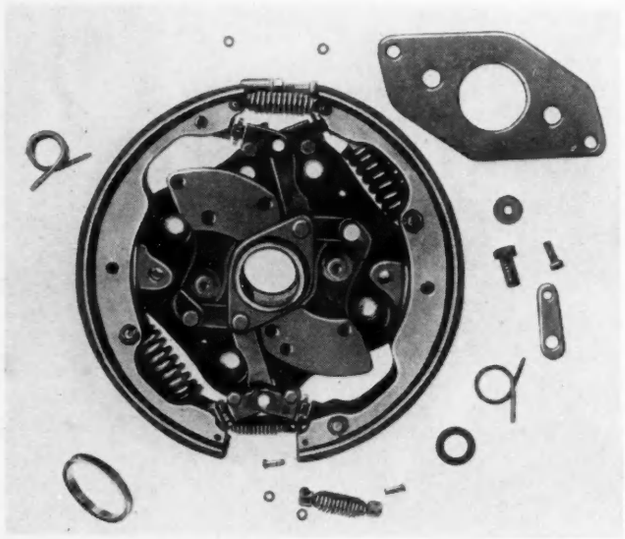

Fig. 5

High Gear

When the accelerator is released for the change from second to third speed, the coasting torque of the vehicle drives the planet housing in the direction of engine rotation as it decelerates. This reversed torque and the speeding up of the planet housing in the direction of engine rotation permitted by the reaction clutch, results in the "throw-out" of the high-speed

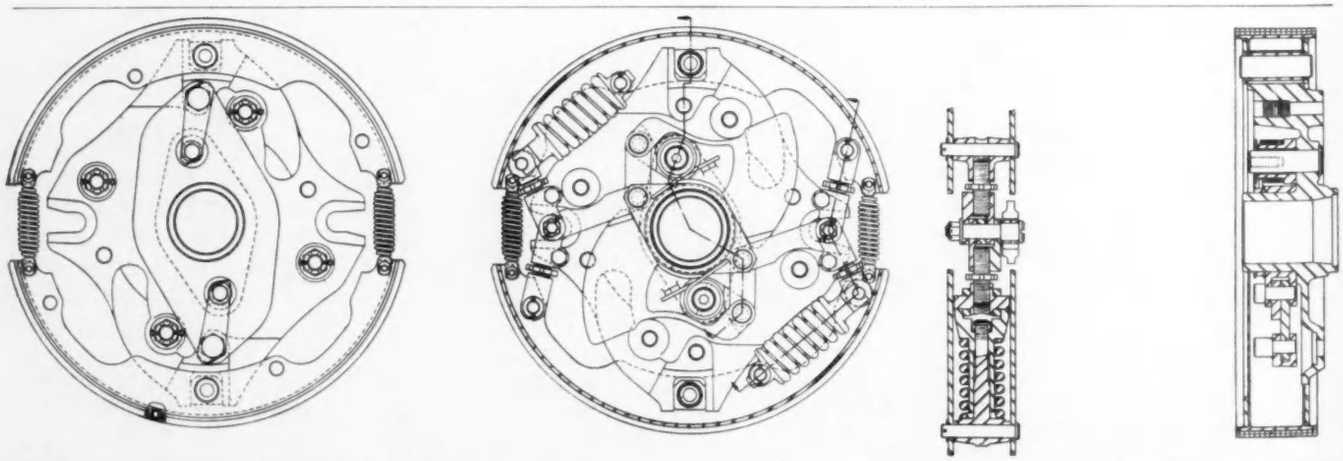

Fig. 6

Figs. 5 and 6 – Mono-Drive Automatic Centrifugal Clutch

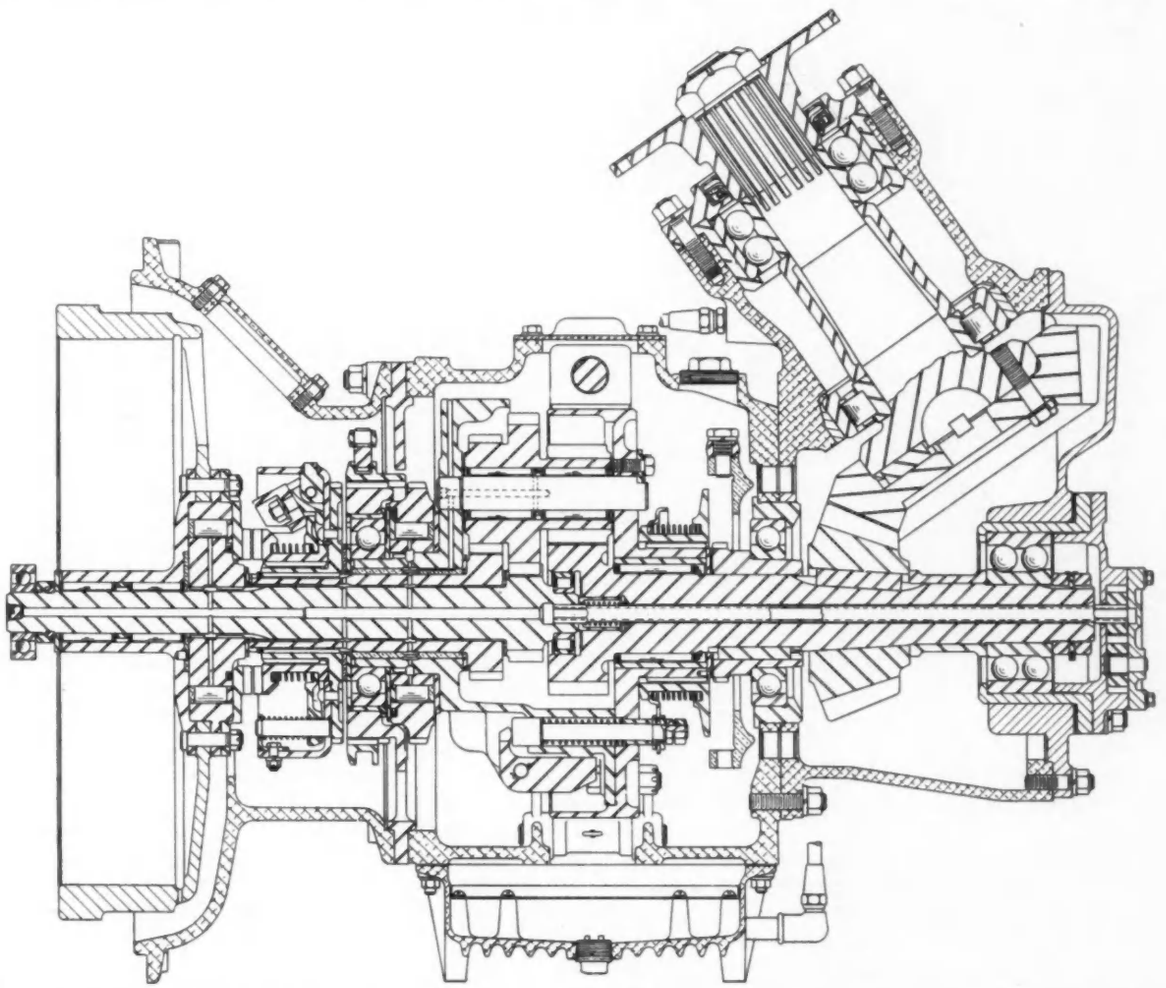

Fig. 7—Cross-Section of Mono-Drive Transmission Incorporating Angle Drive to Axle

centrifugal weights located between the planet clusters and within the planet housing drum. The high-speed sliding clutch is moved back by the radial arm on each of the weights which bear against three slidable studs and a plate contacting with an intermediate coil spring. A gear clutch member keyed to the main shaft takes the drive as soon as both members become synchronized through the change in relative speeds between the planet housing and the main shaft. Upon acceleration of the engine and the engagement of the main clutch, direct drive ensues. The high-speed clutch is shown in Fig. 9. It should be noted that, with the torque reversal, the first-speed shaft attempts to brake the engine; the second-speed shaft is already coupled to it, thus giving a locked system at the time that the high-speed clutch is attempting to engage and while the engine is decelerating and, as there is no relative movement between the two high-speed clutch members, no clashing can occur.

Planet Brake

A pair of brake shoes are hinged at the bottom of the transmission housing as shown in Fig. 10 and are actuated by the air chamber shown in this figure and Fig. 11. The function of the brake is to stop rotation of the planet housing when shifting from "neutral" to "reverse" or "forward" or when using the engine as a brake in second gear.

A control button on the floor of the driver's compartment at the left of the steering column operates this brake. When the transmission is in neutral and the engine being warmed up, the planetary housing usually turns slowly due to a slight drag of the main clutch. The application of the brake permits the initial engagement of the control elements without any clashing and overcomes any inertia of the clutch drum and the mechanism behind it.

Reverse

To initiate reverse speed, the sliding clutch member on the planetary carrier plate is moved to the right to engage a gear clutch member riveted to the back of the second-speed clutch drum, as seen in Fig. 8. As the engine is accelerated, the main clutch comes into engagement and power is conveyed through the roller clutch to the first-speed sun gear. The second-speed sun gear being locked stationary, the planet gear clusters "walk" around the sun gears carrying the planetary housing in a direction opposite to that of engine rotation and with them the mainshaft sun gear at reduced speed.

High-Speed Throw-Out Shoe

To the right of the high-speed clutch is located a throw-out shoe, shown in Fig. 12. It operates only when shifting from third to second or while the transmission is in neutral or reverse. It is actuated by the transmission control lever in the driver's compartment or by the control button. Its purpose, when shifting from third to second, is to push the high-speed clutch forward out of engagement and, when in reverse or neutral (when planet housing is rotating in reverse speed or during warming up), it prevents the high-speed clutch from being moved back into engagement with the main shaft clutch member.

An oil pump is located at the extreme right end and is driven by a long hollow shaft from the first-speed shaft. Oil is conveyed through it to the rifle-drilled first-speed shaft and de-

livered to all transmission bearing surfaces through drilled holes at various points in the mechanism.

Control Lever

The control lever in the driver's compartment is connected through rods and bell cranks to a lever keyed to the shaft on which is mounted the shifting yoke controlling the sliding clutch member on the planet carrier plate and establishing the positions of "neutral," "forward," or "reverse." A cam is mounted on this shaft having three notches in which a poppet plunger engages. The other side of the cam controls the roller of an interlocking push rod which is interconnected with the air-chamber as shown in Fig. 10. During the neutral and reverse positions the interlock is pushed to the right, moving the high speed throw-out shoe to the left. Only in forward speed does the interlock move to the left and permit the shoe to clear the high-speed clutch. The air-chamber and its cam can, at any time operated, throw the shoe to the left.

The individual small units in the transmission lend themselves to economical manufacturing. Only two large-diameter radial ball-bearings are used and, on a basis of equal quantity production, there is no reason to believe that the unit should cost little or nothing over the conventional transmission, clutch, and operating mechanism. The transmission is about 5 in. shorter than the one it replaces and fully 100 lb. lighter.

Operating Speeds

From the description of the transmission it is seen that, in ordinary driving, shifting is accomplished automatically by operating the accelerator pedal. While ordinarily the driver would initiate the shifting to "second" at around 8 m.p.h. and into "high" at 14 m.p.h., there are no restrictions as to when to make the shift. It is entirely feasible to do this at any speed at all, up to the engine-governed speed of 2200 r.p.m. When the bus slows down below 10 m.p.h., the transmission automatically returns to second gear and, below 5 m.p.h., first gear is engaged.

Main Clutch Characteristics

The main clutch drum will not heat up if there is no slippage. It is made of brake-drum cast iron and is painted on the outer surfaces with red lacquer paint that is heat-sensitive and which would shrivel up when the temperature exceeds 300 deg. fahr. In all the operation, with proper linings, none of the drums have shown a discoloration of the paint.

The linings must be hard to withstand extreme pressures and not squeeze out; they must have a high coefficient of friction and be oil- and heat-resistant. The linings used have a dry coefficient of 0.35 and in oil of 0.10. In the initial operation, for experimental purposes, three different makes of linings were used. One proved entirely unsuitable and had to be abandoned quickly, undoubtedly due to the manufacturer's improper analysis of service requirements. Another lining proved to be fair, whereas the third lining now used (a standard woven and compressed type with brass wire mesh) is entirely satisfactory, since after 25,000 miles on the first coach the lining had worn and/or compressed only 0.005 in. The drum was only polished and showed no wear. Of the 29 coaches on the South Division, Nos. 153, 154, 155, 156, 166, 167, 171, 172, and 174, have never been serviced for the transmission or main clutch. No. 154 had one clutch relining for experimental purposes to try a new type (at 2,832 miles) and on Jan. 29, 1937, at 23,691 miles it had never been touched, and the paint on the drum showed no signs of heat. Clutch relining on the other divisions has amounted to only a few cases, due to soft or improper linings in the original vehicles. As in the case of the South Division, about 30 per cent of the coaches

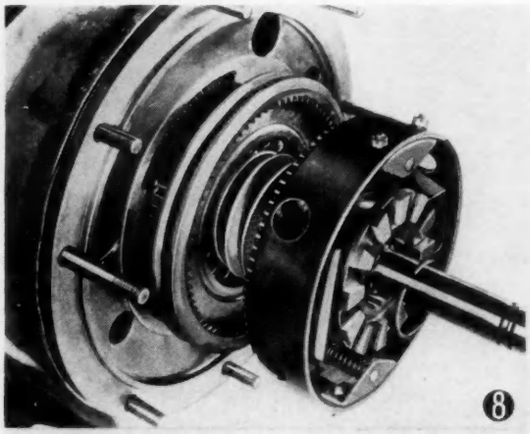

Fig. 8 – Second-Speed Automatic Clutch and Member to Control Planetary Case or Reverse Gear Functions

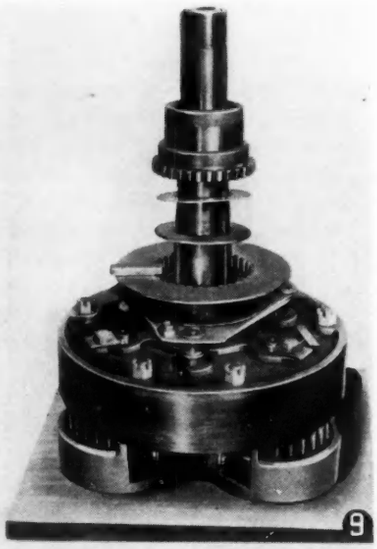

Fig. 9 – High-Speed Clutch and Planetary Case

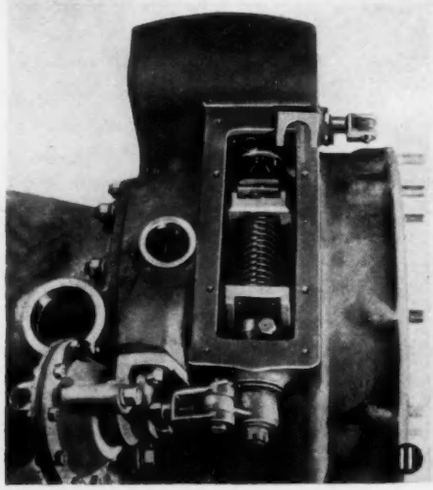

Fig. 11 – View of Air Chamber Controlling Brake Shoes Surrounding Planet Housing and Actuated by the Air Chamber Shown

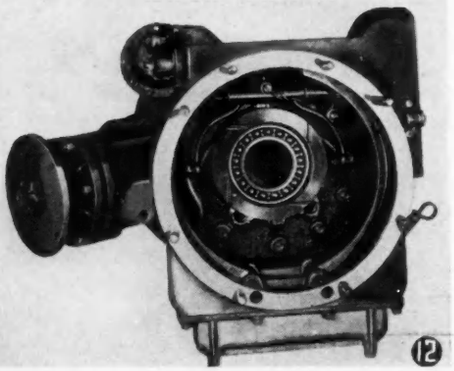

Fig. 12 – High-Speed Clutch Throw-Out Shoe and Angle Drive

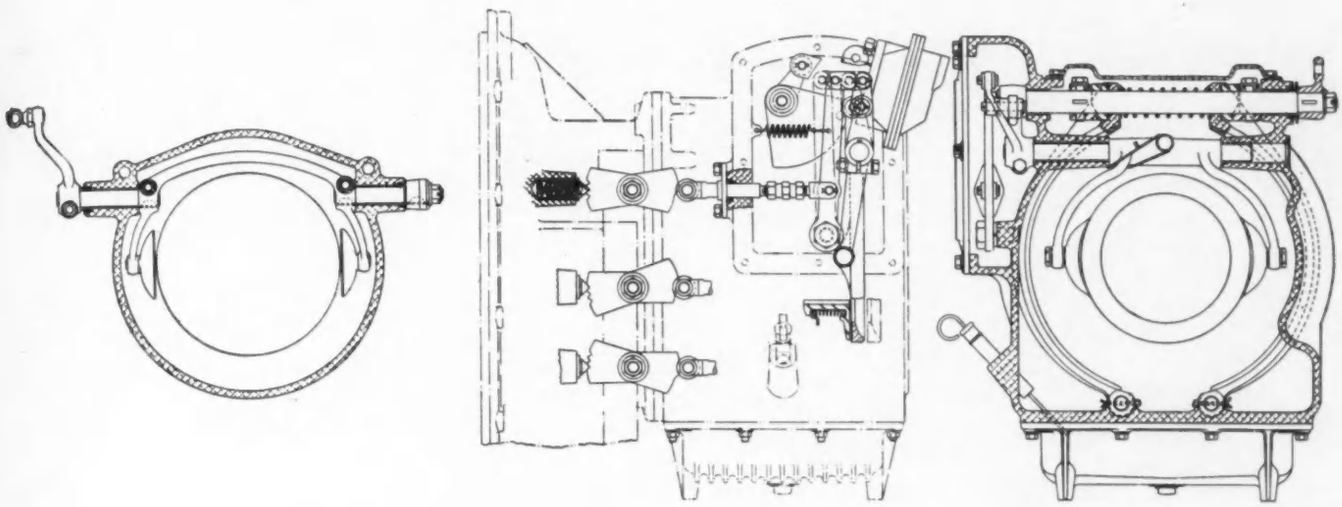

Fig. 10 – Control Members on Transmission

on the other divisions have never been serviced. The servicing of the transmission proper for the entire operation has been negligible. Periodic inspection for the entire vehicle is made every 3000 miles.

The satisfactory clutch performance is due largely to the construction of the unit and the principles embodied therein. The friction surface is at the maximum radius of the clutch as contrasted to the mean radius of the disc type. Tearing action, due to differential slipping speeds, is thus avoided. The forged weights weigh only 4 lb. each for their effective mass. They act through an increasing radius of gyration and the double toggle mechanism, and are then reinforced by the pressure springs. Due to this construction they are able, at 800 r.p.m., to transmit the full engine torque amounting to approximately 450 ft-lb. at this speed. The clutch coverplate gives access to the four star wheels in the secondary toggles, the only points of adjustment in the clutch. As the toggle always stays in the same position, there is no need of adjustment unless there is appreciable lining wear such as would affect the extension of the preloaded pressure spring.

Transmission Performance

During the design period, the greatest criticism was directed at the roller clutches. They have been found to be absolutely dependable, and any perceptible wear has yet to be found.

The smoothness of operation and the entire elimination of any shock load are largely instrumental in effecting trouble-free performance of the gears, shafts, clutches (friction, sliding, and roller) over the entire period, indicating further that no problems over ordinary manufacturing and heat-treatment have arisen. The high-speed toothed clutch was another point wherein doubt was expressed as to its ability to "stay put." No signs of wear are seen at this point, due to the shifting when the relative speed of the mating parts is zero. Furthermore all shifts occur at zero torque.

The disrepute of the planetary transmission in this country is naturally unwarranted as indicated by its great success and use in England. The quietness of the gears in the Mono-Drive is exceptional in view of the use of spur gears. They eliminate any side-thrust problem. The load distribution on the teeth is far more effective than in the parallel-shaft type of transmission, and the distance between supports is very short. The gears have shown no perceptible wear in view of the low tooth loading and elimination of shock. Silence is also due to the design of the cluster gear unit in which each of the three gears has a tooth on a common centerline and the number of teeth in the sun gears are divisible by the number of planets (in this case, 3). The teeth on the second-speed sun gear and the mating cluster gears are 8-pitch and all other gears are 6-pitch.

Lubrication

The transmission does not require any special lubrication, S.A.E. 50 being used for summer and S.A.E. 30 for winter. The lubricant circulates through the planetary system, and some reaches the clutch and lubricates the toggle mechanism. It is then thrown off by centrifugal force around the clutch housing and is guided back to the main reservoir. The circulation of this oil through the clutch no doubt has a cooling effect. If the clutch linings are soft or low in coefficient of friction, the oil temperatures are liable to go quite high, the drums heat up, and the oil cakes on them. Hence the desirability of using the proper type of lining which completely avoids any such contingency. Even if such trouble does occur, the transmission is not affected in any way, and the vehicle can always return to the garage. When the outside temperature was 85 deg. fahr. during a check run last August, the transmission oil temperature ran between 190 and 210 deg. fahr.

In view of the use of a 707 cu. in. engine and the large size and capacity of the coach, it is considered that the mileage obtained is very good. The smooth performance of the transmission was demonstrated very quickly on the Stout Railplane which has a 14-in. clutch behind a 500-ft-lb. torque engine. The Railplane has operated for 35,000 miles with no service on the clutch or transmission with the exception of retraction-spring replacements on the clutch.

In view of the experiences extending over two million miles of operation, it is believed justifiable to state that a new standard of simplified control, smoothness of operation, steady performance, low maintenance costs, and rider appeal has been obtained at the least expense of complication, weight, and manufacturing cost. Any maintenance personnel can take care of these units as no special skill or technique is required in their servicing. A fleet of 100 of these coaches is now operating in New York City and any claims made in this paper that might seem exaggerated can be verified quickly by riding in one of them, either in Chicago or New York.

I am greatly indebted to the Chicago Motor Coach Co., for the data and information that they have made available.

Correcting Diesel Performance to Standard Atmospheric Conditions

By C. Fayette Taylor

Professor of Automotive Engineering, Massachusetts Institute of Technology

THIS paper refers to previous literature on the subject and discusses the fundamental factors involved. It also gives the results of tests showing the effect of variations in atmospheric pressure and temperature on one particular engine. In conclusion it points out the need for further research on this subject.

THE effect of atmospheric variations on the performance of any internal-combustion engine is very complex. A list of the variables involved would be long and would include not only the obvious variation in the quantity of air available for combustion, but also variations in fuel-air ratio, in the process of combustion, in mechanical and pumping friction, in volumetric efficiency, and so on. In spite of this situation, the behavior of Otto-cycle engines is sufficiently consistent so that moderately satisfactory empirical formulas can be used for correcting the performance of conventional engines of this type for the normal variations in atmospheric conditions which occur at any one altitude. Less satisfactory – although still usable – rules are available for predicting the performance of the conventional Otto-cycle engine over the range of altitudes encountered in aeronautical service.

The problem of correcting Diesel-engine performance for atmospheric variations is more difficult than in the case of the Otto-cycle engine, chiefly on account of the fact that Diesel engines are operated over a very wide range of fuel-air ratios whereas, for Otto-cycle engines, the fuel-air ratio is held at, or near to, a fixed value. Also, in the case of Diesel engines, atmospheric changes tend to alter the fuel-air ratio because such engines generally are operated at a fuel rate fixed by the adjustment of a mechanical injection system. Changes in air density are not necessarily, in fact not usually, accompanied by changes in the fuel rate, and hence the fuel-air ratio varies inversely as the density of the air taken into the cylinders. In the case of the Otto-cycle engine, there is at least partial correction of the fuel rate due to inherent carburetor characteristics, and furthermore, the ratio of fuel to air in the Otto-cycle engine is usually in the region where small changes have very little effect on engine performance.

Another factor that makes the problem more difficult is that the combustion process in the compression-ignition engine is more sensitive to atmospheric changes than in the Otto-cycle engine, unless the latter is on the verge of detonation. Although large changes in altitude do affect the Otto-cycle combustion rate, the normal variations which occur at a constant altitude do not seem to have an appreciable effect. In the case of the compression-ignition engine, even small changes in inlet pressure and temperature may have appreciable effects on fuel distribution, on fuel evaporation, on the delay period, and on the subsequent rate of combustion. Fig. 1 shows indicator diagrams taken on the M.I.T. sleeve-valve Diesel[1] at various inlet pressures. Boerlage and Broeze[2] have shown the influence of varying inlet temperature on the delay period, and Rothrock[3] has shown the effect of variations in the air temperature on fuel evaporation and on combustion rate. The effect of such changes in the process of combustion on engine performance would be minimized if the fuel-injection timing were altered to bring the peak pressure at the same point in the stroke. In practice, however, this adjustment is seldom done in response to minor changes in atmospheric conditions, with the result that the angle at which the peak pressure occurs changes with variations in inlet temperature and pressure.

Experimental Work

In order to add something to the rather limited amount of experimental work which has been done in this field to date[4,5], the effects of variations in inlet pressure and temperature on engine output were determined for a single-cylinder, sleeve-valve, compression-ignition engine in the Sloan Automotive Laboratory of the Massachusetts Institute of Technology. The engine unit is described in a paper published in *Diesel Power*[1], and a photograph of the test installation is shown in Fig. 2. The engine was connected to an electric cradle dynamometer and equipped with the usual arrange-

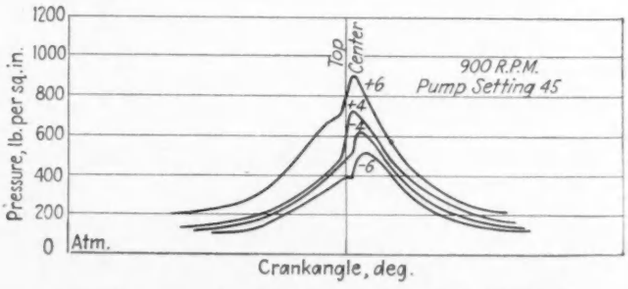

Fig. 1 – Pressure-Time Indicator Diagrams with Various Inlet Pressures – M.I.T. Sleeve-Valve Compression-Ignition Engine

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1937.]

[1] See *Diesel Power*, September, 1933, pp. 550-553; "Compression-Ignition Tests with a Sleeve-Valve Engine," by C. F. Taylor and M. S. Huckle.

[2] See S.A.E. TRANSACTIONS, July, 1932, pp. 283-293; "Ignition Quality of Diesel Fuels as Expressed in Cetene Numbers," by G. D. Boerlage and J. J. Broeze.

[3] See N.A.C.A. Technical Reports: No. 435, 1932, by A. M. Rothrock and C. D. Waldron; No. 512, 1935, by A. M. Rothrock and Mildred Cohn; and No. 525, 1935, by A. M. Rothrock and C. D. Waldron.

[4] See *Automotive Industries*, Sept. 9, 1933, pp. 303-304; abstract of paper "Reducing the Performance of a Solid-Injection Diesel Engine to Standard Conditions," by H. A. Everett, presented at the Oil Power Conference, 1933.

[5] See S.A.E. TRANSACTIONS, June, 1935, pp. 210-214, 236; "A Rational Basis for Comparing Diesel-Engine Performances," by E. S. Dennison.

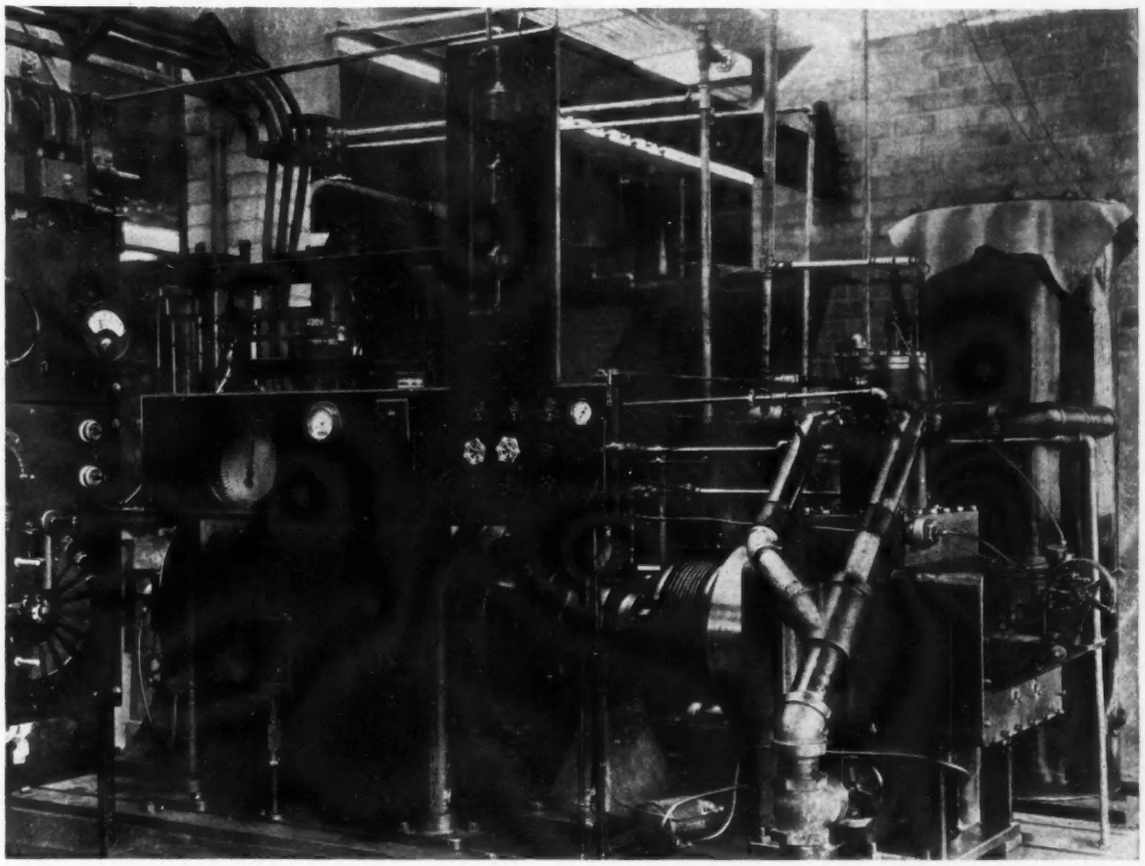

Fig. 2 – M.I.T. Sleeve - Valve Compression - Ignition Engine on Electric Dynamometer

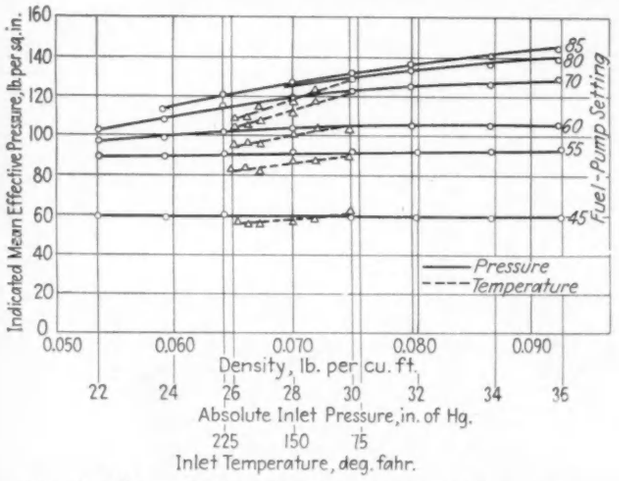

Fig. 3 – Effect of Inlet Pressure and Inlet Temperature on the Indicated Mean Effective Pressure at Various Fuel-Pump Settings – M.I.T. Sleeve-Valve Compression-Ignition Engine at 900 R.P.M.

ments for measuring torque, revolutions per minute, fuel rate, inlet pressure, inlet temperature, and so on. An electric heater for the inlet air was mounted in the inlet system. Temperature variations were secured by varying the amount of current supplied to this heater. Pressure variations were secured by throttling the inlet air, or supercharging by means of a separately driven air pump. The range of inlet pressure explored was from 22 to 36 in. hg. absolute, and the range of inlet temperatures was from 76 to 215 deg. fahr. Exhaust pressure was atmospheric throughout the tests. The actual test work was conducted by Messrs. Talbert and Randell, students at the Massachusetts Institute of Technology[6].

[6] See "Performance Characteristics of a Compression-Ignition Engine with Varying Conditions of Pressure and Temperature of the Inlet Air," by O. B. Randell and J. E. Talbert, Thesis, June, 1935, on file in the Library of the Massachusetts Institute of Technology.

Methods of Computation

In order to eliminate, as far as possible, the influence of engine friction, the results of the tests are given in terms of indicated mean effective pressure. This pressure was computed by adding to the brake mean effective pressure the "friction mean effective pressure," as determined by motoring with atmospheric inlet and exhaust pressure. This value was corrected further by adding, algebraically, the difference between exhaust and inlet pressure, for the purpose of reducing the results to the condition where exhaust and inlet pressure vary together. This is the usual case where changes in atmospheric conditions are involved. Although this method of correction is open to question, the results obtained are apparently within the limits of accuracy to be expected from the motoring test itself.

Results

Fig. 3 shows the variation of indicated mean effective pressure with variation in inlet density at various rates of fuel flow. It will be noted that the results are not the same when the density is varied by changing the temperature as they are when the density is varied by changing the pressure. This condition is true also in the case of the Otto-cycle engine.

Perhaps the most significant trend shown in Fig. 3 is the fact that the effect of pressure and temperature is relatively small with a low fuel rate (large amount of excess air) and that the effect of both temperature and pressure increases as the fuel rate is increased. This trend is to be expected. When there is a large amount of excess air, the energy of the fuel-air mixture varies almost directly with the quantity of fuel. If this quantity is held constant by a constant fuel-pump setting, the effect of pressure and temperature changes on the indicated mean effective pressure is small. On the other hand, when the quantity of fuel injected is sufficiently large so that there is little or no excess air with normal atmospheric conditions, throttling produces excessively rich mixtures with consequent poor combustion, and supercharging tends to

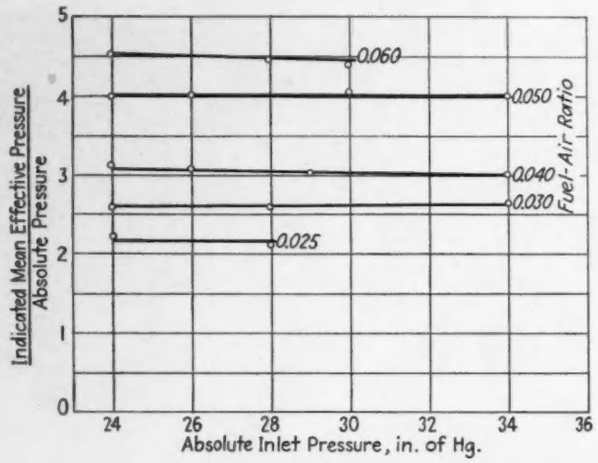

Fig. 4 – Effect of Inlet Pressure on the Ratio of Indicated Mean Effective Pressure to Inlet Pressure at Various Fuel-Air Ratios – M.I.T. Sleeve-Valve Compression-Ignition Engine, 900 R.P.M.

supply the necessary excess air for more satisfactory combustion conditions and more complete utilization of the fuel. It is not surprising, therefore, that, with high fuel-air ratios, there should be rapid variation in the engine output with changes in the inlet pressure or temperature. The fact that a change in inlet temperature produces a greater effect on indicated mean effective pressure than when the same change in density is effected by a change in pressure, is probably due to the stronger influence of temperature on combustion.

In view of the importance of the fuel-air ratio in determining the behavior of the engine under variations of inlet pressure and temperature, Figs. 4 and 5 have been plotted from the original data to show curves of constant fuel-air ratio.

In the case of the Otto-cycle engine, the indicated mean effective pressure is nearly proportional to the absolute inlet pressure. This being the case, the indicated mean effective pressure divided by the absolute inlet pressure should be a horizontal straight line when plotted against the inlet pressure. Fig. 4 is this type of plot for the compression-ignition engine in question at various fuel-air ratios. At a given fuel-air ratio the compression-ignition engine would be expected to behave in a manner more nearly similar to that of the Otto-cycle engine, and the curves certainly show a tendency in this direction. The departure from the horizontal straight line, for any fuel-air ratio, is not much in excess of the experimental error involved.

If it is assumed that the indicated mean effective pressure varies directly with the air density, the product of the indicated mean effective pressure times the absolute temperature should

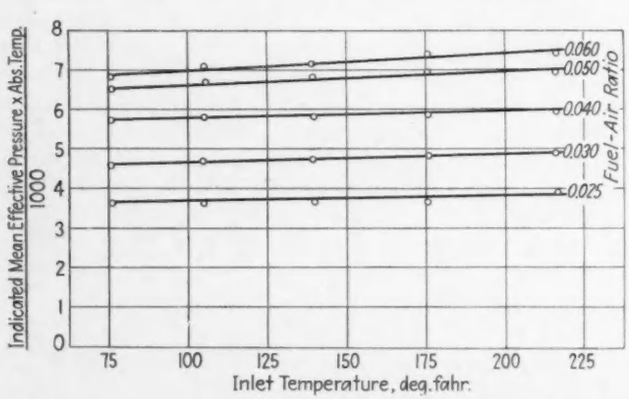

Fig. 5 – Effect of Inlet Temperature on the Product of Indicated Mean Effective Pressure Times Inlet Temperature at Various Fuel-Air Ratios – M.I.T. Sleeve-Valve Compression-Ignition Engine, 900 R.P.M.

be constant. In Fig. 5 this product is plotted against the inlet temperature for various fuel-air ratios. In this case there is a definite tendency for the curves to slope upward to the right. This slope may be due to improved volumetric efficiency or better "timing" of the peak pressure, or both.

In considering such results as those shown herewith, it should be remembered that the change in performance with a given change in inlet conditions depends to a considerable extent upon the particular injection timing used. For example, if the timing is intentionally set "late" (as is often the case in order to control maximum pressure), an increase in pressure or temperature will have the effect of improving the performance by reducing the delay, entirely apart from any other effect. On the other hand, if the timing of injection is that for best power, or earlier, the advance in effective timing due to an increase in inlet pressure or temperature will be detrimental.

Conclusions

It is quite evident that the magnitude of the effects of inlet pressure and temperature changes on the power output of a compression-ignition engine will depend on particular conditions of design, operation, and adjustment. However, it is believed that the following trends, shown in the case of one particular engine, are generally characteristic of the so-called "Diesel" engine.

(1) The effect of a given change in atmospheric pressure or temperature on engine power is small with a low fuel-air ratio (large quantity of excess air) and increases as the fuel-air ratio increases.

(2) At a constant fuel-air ratio, the performance varies in a manner generally similar to that of the Otto-cycle engine.

There would seem to be some hope of developing approximate empirical formulas for the correction of Diesel-engine performance to standard atmospheric conditions at a given fuel-air ratio. Such formulas would have to be based on tests of a large number of different kinds of compression-ignition engines. But the development of a correction for the condition of constant fuel rate will be difficult, as this factor is affected strongly by particular design and operating conditions.

Die Castings for Automotive Parts

(*Continued from page* 302)

often overlooked as that of consulting the die caster fully, and as early in the design as possible. The capable die caster is usually versatile enough to get around difficulties in casting occasioned by poor design from the casting standpoint, but he naturally must charge for the extra work and die expense involved. The die caster thinks first of such matters as gating, which have much to do with producing perfect castings of good finish, whereas the designer looks more toward the results to be attained by the casting when he gets it. The objective is usually to get the best part at the lowest cost consistent with desired results in the application of the casting, and there are rarely times when early and frequent consultation with the die caster will not help to secure this result. It need hardly be added, perhaps, that the metal suppliers are always ready to assist in such matters as fall within their province and to give their opinions, after suitable research if it is required, as to performance under given conditions, especially when these conditions are unusual. There is, of course, a great wealth of accumulated experience with die castings with which both the metal supplier and the competent die caster are familiar, and either is ready to share his experience with those having a legitimate interest in profiting by it.

Lubrication and Cooling Problems of Aircraft Engines

By Weldon Worth

Assistant Mechanical Engineer, U. S. Army Air Corps

THIS paper presents the problem of engine lubrication from an installation and operation standpoint with respect to starting, warm-up, and stabilized flight. It describes the manner in which these problems are solved by the latest Materiel Division lubrication system, with its oil dilution, hopper-type oil-tank, viscosity-control valve and jacketed oil coolers.

It also analyzes the radiator drag and shows the magnitude of unnecessary radiator drag that may exist if the radiator is not properly designed and installed. The paper then presents a practical design method and shows a sample design calculation for a 1000-hp. engine.

THIS paper starts with the assumption that all engine-design problems of lubrication and cooling have been solved satisfactorily and proceeds with the problem of keeping them solved when the engine is installed in an airplane and expected to perform under the full range of the varying conditions of heat and cold, of rain and snow, of altitude, attitude, speed, accelerations, and vibration. Where else will one find this range of operating conditions? But the airplane in a few hours may run the full gauntlet of these conditions, and the lubrication and cooling systems must operate successfully and efficiently. They must provide lubrication and temperature control that will permit engine operation in the airplane that is consistent with dynamometer performance.

Lubrication Systems

The problem of the lubrication system, from on operating standpoint, may be broken down into three subdivisions:

(1) That of supplying lubricating oil to the engine in the proper condition of temperature and viscosity under the stabilized conditions. This work must be done for long periods of operation at full power or at cruising power; in level flight or climb; and throughout the full range of air speeds, temperature, and other varying conditions.

(2) That of obtaining a lubricant condition in cold weather which permits starting the engine, rather than a condition where the lubricant is so stiff that the engine cannot be turned over at the cranking speeds necessary to permit starting. Certainly if the oil is so stiff and viscous that the engine cannot be started, the ingenious solution of all previous airplane and engine problems has gone for naught.

(3) That of supplying good lubrication to the engine immediately after starting and the elimination of an undesirable oil-warm-up period. During this period most pilots have watched the oil pressure gage fluctuate and hoped (or prayed) that the friction surfaces would overlook any temporary lapse of lubrication.

Continuous Operation

The first phase of the problem covering operation for extended periods not only requires oil cooling but the cooling must be controlled to maintain the oil at a viscosity where it will circulate freely and lubricate properly.

This control requires an oil cooler but offers a choice between shutters to control the air flow or a bypass valve which will direct the flow of oil through or around the cooling elements. The valve has been selected for Air Corps use and is preferred to shutters chiefly because of its susceptibility to automatic control. It thus avoids both the need for attention during flight and the consequences of neglecting this attention. The valve is also lighter in weight and requires less maintenance than shutters.

With an unjacketed type of oil cooler, when cold weather is encountered, the oil may bypass the core completely for a while to avoid over-cooling. When this condition occurs, the oil will congeal in the core and, if the cooling air is cold enough, circulation cannot be reestablished. In this case that peculiar condition exists where the oil to the engine is too hot because the cooler is too cold.

To avoid this condition the jacket is added to the cooler, and it surrounds the entire unit so that all the bypassed oil flows through the jacket and, in so doing, warms a complete passage through the core. This arrangement prevents complete congealing in the core and permits the reestablishment of the oil flow when necessary for cooling.

The two paths through the cooler are illustrated in Fig. 1. The bypass path that is used when the oil is cold is indicated by the solid line, and the cooling path when the oil is hot is indicated by the dotted line.

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., Mar. 12, 1937.]

This type of cooler is reasonably efficient and uses the ordinary cartridge type of core with a well baffled oil flow. Air efficiency of 75-80 per cent is obtained with oil compared to 100 per cent for the same surface when flowing water at similar air flows and temperatures.

The use of the jacket makes it possible to control the cooling in all conditions of flight with a regulating valve. This possibility brings up the problem of a valve to control the cooling.

The lubrication of the engine is a function of the viscosity of the oil rather than of the temperature. Since a temperature control is correct only for one grade of oil, it is basically more desirable to have the valve function on a viscosity rather than a temperature index. In other words, it is desirable to cool a thin or diluted oil to a lower temperature than a heavy oil because some given viscosity is more desirable for circulation and lubrication. The design of a valve with the desired characteristics has been quite a problem, and the Materiel Division has spent several years of study and cooperation with manufacturers in the development of the present valve. The Type F-4 viscosity valve, illustrated in Fig. 2, is now on Service Test Status. By regulating the oil flow to the core of the cooler, it controls the amount of oil cooling and tends to hold the oil to a constant viscosity within the limit of the cooling capacity of the cooler. In addition to this advantage

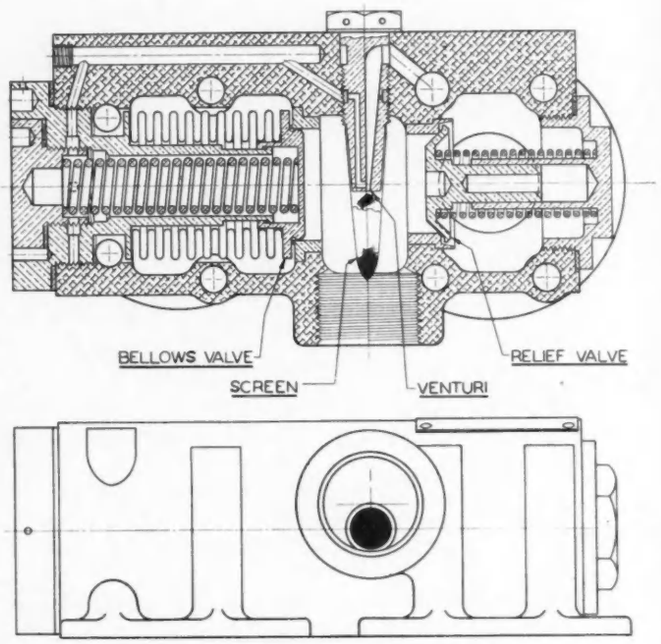

Fig. 2 – Viscosity-Controlled Bypass Relief Valve – Type F-4

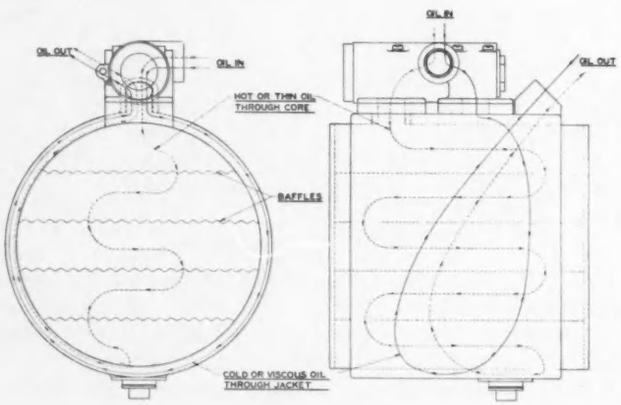

Fig. 1 – Jacketed Oil Cooler with Viscosity Valve

of viscosity regulation, a viscosity valve may be preferable to a thermostatic valve because it is free from the effects of altitude and will go to a cooling position in case of failure of the flexible metal bellows. It also should go to a cooling position in the case of the other most probable types of failures such as clogging of the screen or venturi. To date no such failures have occurred, however, and tests have indicated that, with the small oil flow through the screen and the high velocity past it, any foreign matter either passes through without difficulty or is washed away from the screen.

The viscosity control F-4 valve, illustrated in Fig. 2, attaches directly to the cooler and requires no external plumbing. The inlet oil to the cooler passes into the valve inlet from which it may choose one of the two paths. When cold or viscous, it passes through the relief valve and around the oil cooler jacket as illustrated in Fig. 1. When the oil is hot and thin, the pressure in the bellows valve is reduced until it opens, and the relief valve then establishes a maximum pressure to force the oil through the bellows valve and the cooling elements of the cooler. Thus, the operation of the valve is controlled by the pressure in the bellows. This pressure is controlled by the throat pressure of a small venturi that bypasses a very small quantity of oil from the inlet to the low-pressure side of the relief valve. The effect of viscosity upon the flow and pressure furnishes the necessary pressure variation to operate the valve.

Lubrication at Starting

We will next proceed to the second subdivision which deals with the difficulty of starting the engine in cold weather. The present design of aircraft engines apparently needs a lubricating oil of a high viscosity. At the normal maximum oil-inlet temperature of 185 deg. fahr., the standard grade of oil has a Saybolt viscosity of 185 sec. as compared with 70 sec. for ordinary automobile oil, S.A.E. No. 20. Even the standard grade of winter oil used by the Air Corps has a viscosity of about 350,000 Saybolt sec. at 0 deg. fahr., whereas the S.A.E. No. 20 oil has a Saybolt viscosity of about 6 per cent of that figure. When these aircraft oils are lowered to the temperatures encountered in winter operations away from heated hangars, it is impossible to turn the engine over at a reasonable cranking speed with any means ordinarily available, and special steps must be taken. The usual methods employed are fire pots, heating hoods and so on, which, in all cases, are bulky and cumbersome. They are usually slow in operation and, in many cases, they introduce an appreciable fire hazard.

Some dual oil systems have been considered with the separate circulation of a light oil for starting, necessitating double tanks, double oil servicing, and a system of valves which, if neglected, will wreck an engine. The best solution at this time seems to be the oil dilution system that is now undergoing extensive tests by the Air Corps. It was very successful during experimental development and during flight tests that were conducted in the winter of 1935-36 and 1936-37 by the Materiel Division at Dayton, and on flights to localities of colder weather.

This system is based upon the established principle that, within practical limits, a cold oil, when diluted to the proper lubricating viscosity by the addition of gasoline, will provide satisfactory lubrication. Certainly the grease-like viscosity of ordinary oil at sub-zero temperatures is not only a poor lubricant that fails in its basic purpose of reducing friction, but

it actually prevents the relative motion of the engine parts when cranking is attempted.

The oil dilution system with a total weight of a very few pounds provides for thinning the circulating engine oil, prior to stopping. This thinning will permit the engine to be started upon a moment's notice after exposure to sub-zero temperatures for long periods of time. This system, of course, assumes, as before stated, that the carburetion, priming, ignition, and other accessories are functioning properly. When diluting the oil in this manner, no auxiliary apparatus or heat of any kind is used, and the complete equipment is carried in the planes at all times. In tests last winter the airplane was placed on the line with no engine covers or protection from the cold, and the start was made by the pilot without the assistance of mechanics or other personnel. After leaving the plane outside overnight, a test was made the next morning with a temperature of —10 deg. fahr. It required less than 4 min. from the time the pilot approached the airplane until a take-off was completed. If necessary, it is believed that the time can be reduced further. With the latest refinements in the other accessories, no difficulties have been experienced in obtaining starts at the coldest temperatures that have been encountered, namely, —20 deg. fahr.

Lubrication After Starting

This section reveals in part the solution to the third phase of the problem dealing with proper lubrication immediately after starting, and with the reduction of the warm-up period. It is obvious that, if the cold oil is at the proper lubricating viscosity by reason of its dilution, the lubrication of the engine is satisfactory at time of starting and there is no need for long warm-up period.

Elements of System

To clarify the relation of the special lubrication system to the second and third phase of the problem, the elements of the system and their operation will be explained. The dilution of the oil is accomplished by a connecting line between the fuel pressure line and the oil line entering the engine as shown in Fig. 3. In this conneeting line are located a metering unit and a shut-off valve that is operated from the cockpit. The valve has a spring return to the "off" position so that it is open only when held in that position. This is the only control or movable part in the system and there is, therefore, no danger from neglect in operation. The only act performed in the servicing or operating the system is holding this valve open for 3 or 4 min. at the time when the engine is shut down when a start at cold temperatures is anticipated. When this operation is done, the engine can be turned over in sub-zero temperatures as freely as in the summertime.

In the dilution of the oil it is desirable to have the thinned oil circulated thoroughly through the engine, but it is not desirable, however, to have the entire oil supply diluted. To avoid this condition a special arrangement was designed to segregate a small portion of the oil that is circulating through the engine and keep it separated from the main oil supply in the tank. The hopper-type tank accomplishes this result and, at the same time, insures the availability of the main oil supply at all times through unrestricted passages. (Passages *A* in Fig. 3.)

As shown in Fig. 3 the hopper conducts the oil from the inlet directly to the outlet and there is no interflow with the main part of the tank. With the high oil flows and small volume of the hopper (approximately 1 to 1½ gal.), special care must be taken to separate the air from the oil before it is returned to the engine. This air becomes entrained during the scavenging from the crankcase and can be separated best from the oil by a tangential entry as illustrated in Fig. 3. This arrangement directs the oil tangentially and downwardly against the hopper wall in such a way that centrifugal separation may take place. In this arrangement care must be exercised to avoid any turbulence that would interfere with this centrifugal separation of the entrained air.

It is interesting to note Fig. 4 which is one of the early attempts at a hopper-type oil tank that was built with a glass side and tested under observation in the cold room. The hot oil entered and flowed downward through the hopper very nicely but at the bottom it misbehaved and, instead of going to the outlet, the hot oil passed under the partition while the cold heavier oil clung to the bottom and flowed across the tank under the partition and thence to the tank outlet. This flow took place even when the partition extended to within ½ in. of the bottom of the tank. The arrangement shown in Fig. 4 is rather similar in appearance to Fig. 3 but it is useless, while the one in Fig. 3 works with satisfactory results. A study of the two illustrations will explain the difference in

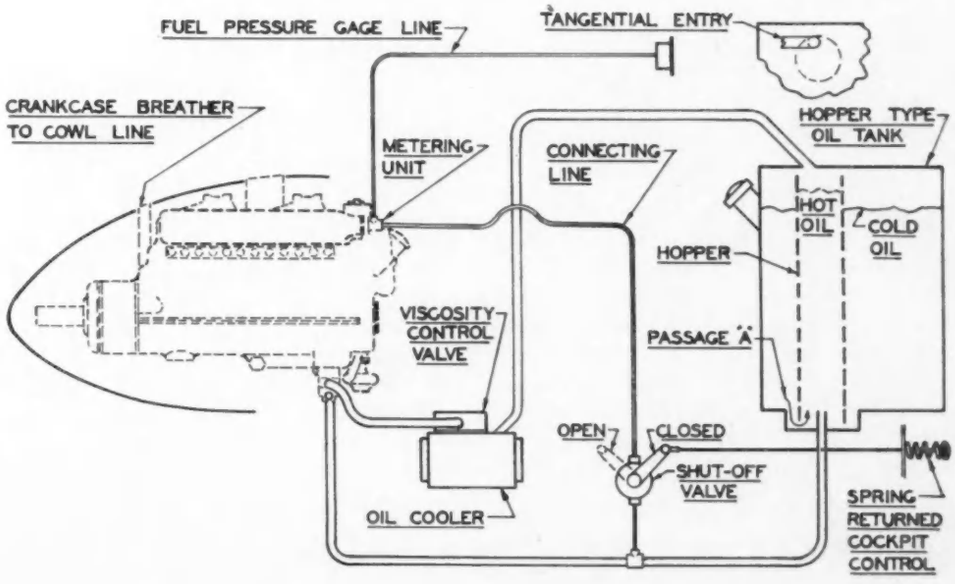

Fig. 3 – Lubrication System with Provision for Oil Dilution

operation if consideration is given to the balance between the liquids of different density in the two compartments and to the flow equilibrium that is established.

The hopper-type tank is considered essential to the system in solving the third phase of the problem along with the other elements to control the viscosity of the oil during the transition from starting to continuous operation. The diluted oil provides starting and immediate lubrication with a cold engine, but, obviously, as the engine and the oil warm up, the dilution is no longer needed, nor is it desirable. The transition from the diluted cold oil to normal undiluted warm oil is well taken care of by two circumstances. The heat of the engine tends to distill the gasoline from the oil as it circulates. In addition to this elimination of gasoline, the circulating oil segregated by the hopper is consumed by the engine and replaced by fresh undiluted oil. With these two factors at work and the additional cooperation of the viscosity valve to control the oil to the proper viscosity regardless of its temperature, a substantially constant lubricant viscosity is maintained during the complete cycle of operation. It requires a period of 15 to 30 min. of engine operation to eliminate all effects of the dilution and from then on the lubrication is entirely normal. The majority of the dilution is eliminated in the first 5 or 10 min.

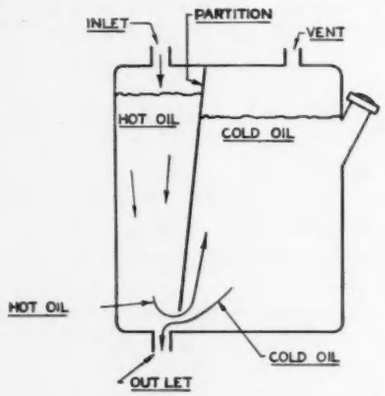

Fig. 4 – Hopper-Type Tank That Failed to Function

This method may raise another point which was considered carefully – the fire hazard that results from the fuel vapors in the crankcase that are discharged from the breathers. Investigation disclosed that, even with the normal system, the vapors discharged from the engine breather will burn with surprising spontaneity. The distillation of gasoline from the oil in the crankcase merely tended to increase the volume of the flame. There is probably an over-rich mixture in the crankcase and it is an opinion that oxygen in the crankcase would be much more dangerous. There have been no difficulties of any kind in this respect but, to be on the safe side, the engine breather is extended to beyond the cowl line in all installations of the oil dilution system.

Aside from the operation of the hopper-type oil tank with respect to the dilution it has two other functions: It provides an extremely rapid warm-up by reducing the quantity of oil that must be heated to approximately 1½ gal. This provision reduces the time required to reach a temperature equilibrium from what has actually been hours in some cases, to a few minutes. The other function of the hopper-type oil tank is rather startling. It is made with some hesitancy and with the realization that it will not be accepted without a storm of protest from some sources. In my opinion an analysis of the operation and the actual tests indicates that there is no reason to drain the oil between overhauls unless there is an engine failure of some type, and this necessity would be indicated readily by an inspection of the oil-sump screen.

With an all-usage oil system that pumps oil to the engine only as it is consumed, there would be no scavenged oil and, hence, no engine usage or aging of the oil in the tank. The hopper-type oil tank approaches this all-usage system by segregating a small portion of the oil which is recirculated at a high rate until consumed and replaced by fresh oil. With 1 gal. circulating and a consumption of 1 gal. per hr., 99½ per cent of the oil always will have less than 5 hr. of engine operation regardless of the number of hours between oil changes. A test of the system was conducted on a modern two-row radial engine in which the oil was diluted excessively, allowed to rest 1 hr., operated at full throttle for 1 hr., diluted, allowed to rest 1 hr., and so on for 70 hr. of full-throttle operation. With respect to the percent of acidity, solid matter, and carbon deposit, the oil was as good in all respects after 70 hr. of engine operation as it was at 10 hr., and substantially the same as it was after a run-in period of 5 hr. before there was any dilution. The instructions have been issued by the Materiel Division not to change oil in airplanes equipped with the oil dilution system.

Liquid Cooling

The second section of this paper will deal with cooling systems for liquid-cooled engines. With our phenomenal development of air-cooled engines during the past decade, many people, forgetting that they drive a liquid-cooled automobile engine, have concluded that all engines will be or should be air-cooled. To those people who doubt the possibility of liquid-cooled engines, it is suggested that they review the type of engines that holds the two world's maximum speed records for both land and sea planes. In addition, consideration should be given to the outstanding performance of military airplanes being developed here and abroad with liquid-cooled engines. It is believed that where high power, maximum performance, high altitude flying, and low specific fuel consumption are important, the liquid-cooled engine will stage a comeback for certain types of use. H. Wood of Rolls-Royce, Ltd., ably presented the case of the liquid-cooled engine at one of your recent meetings.[1] Europe has developed liquid-cooled engines whereas we have devoted the last decade almost exclusively to air-cooled engines – with remarkable success. However, even in this country, the liquid-cooled engine is gaining a new lease on life.

The Materiel Division always has had faith in the liquid-cooled engine and, as far back as 1931, the Division foresaw a greater use of liquid-cooled engines than existed at that time,

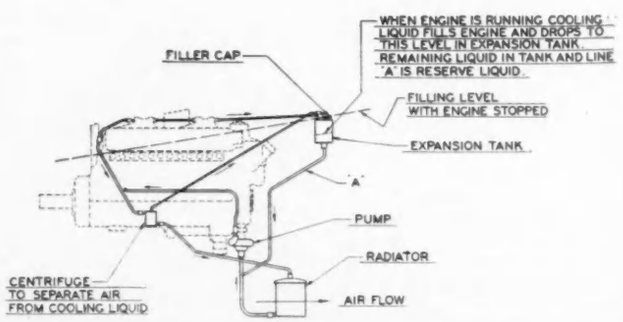
WHEN ENGINE IS RUNNING COOLING LIQUID FILLS ENGINE AND DROPS TO THIS LEVEL IN EXPANSION TANK. REMAINING LIQUID IN TANK AND LINE 'A' IS RESERVE LIQUID
FILLER CAP
FILLING LEVEL WITH ENGINE STOPPED
EXPANSION TANK
'A'
PUMP
RADIATOR
CENTRIFUGE TO SEPARATE AIR FROM COOLING LIQUID
AIR FLOW

Fig. 5 – Standard Air Corps Cooling System

[1] See S.A.E. TRANSACTIONS, July, 1936, pp. 267-287; "Liquid-Cooled Aero Engines," by H. Wood.

and instituted further radiator and cooling-system investigations.

Arrangement of Cooling System

We will first discuss briefly the cooling system installation from the plumbing standpoint. Fig. 5 shows the arrangement that a varied field of experience and testing under operating conditions has proved superior as a whole. Deviations from this system are very unpopular with the Air Corps. It is rather difficult to curb the originality of the engineers in a resourceful field like aeronautics but, so far as the plumbing arrangement is concerned, the difficulties that have been experienced with systems "just as good" provide sufficient basis for this attitude.

The outstanding requirement is the location of the expansion tank with respect to both position and its connection to the system, and the inlets and outlets of the radiator.

The expansion tank controls the absolute pressure in the system since it contains the only vent to the atmosphere. Therefore, if the expansion tank is connected to the lowest pressure point in the system, the pump inlet, all other pressures will be higher and there will be no high suction pressures to cause cavitation or a breakdown in the flow.

The location of the expansion tank and the filler cap at the proper elevation with respect to the engine when the airplane is at rest on the ground solves one problem of servicing. With the required location there is sufficient space in the engine above the filler opening to provide for the expansion of the coolant when it heats up and avoid the loss of liquid. This space in the engine is filled up immediately by coolant flow when the engine is started and the expansion space is transferred to the tank.

With the standard system the Prestone is filled to the level of the filler-opening, which is the easiest level to determine. This arrangement avoids the use of petcocks that may be left open and other devices for determining a filling level.

The coolant outlet from the radiator must connect to the bottom header. This requirement avoids a sudden failure in case of coolant leakage. With the outlet located at the bottom, the inlet to the pump will be drowned as long as there is any coolant in the radiator and, when there is a loss of liquid beyond the reserve supply, the radiator will be only partially filled but circulation will be maintained. This condition will be indicated by a rising temperature. Where the outlet is at the top of the radiator and air from the inlet can pass directly to the outlet, tests indicated that the circulation will break down suddenly and completely upon the loss of slightly more than the reserve capacity.

Some types of liquid-cooled engines are susceptible to leakage of combustion gas from the cylinder into the coolant passages. This leakage may force the liquid out through the overflow and has caused many a cooling-system failure. To avoid this consequence, the Air Corps has equipped all susceptible engines with a centrifuge to separate the combustion gas and avoid these consequences. It has completely eliminated this type of failure.

Radiator Drag

With engine horsepower twice as great and speeds increased by the same ratio the combined increase in radiator-drag horsepower without any improvement of radiator installations would be about 800 per cent. Assuming a 1925 radiator installation with no cowling, the solid curve in Fig. 6 shows the approximate drag horsepower that the uncowled radiator will require to cool a 1000-hp. engine (not available of course in 1925). At 300 m.p.h. this radiator would absorb about 20 per cent of the engine power as drag. In the evaluation of

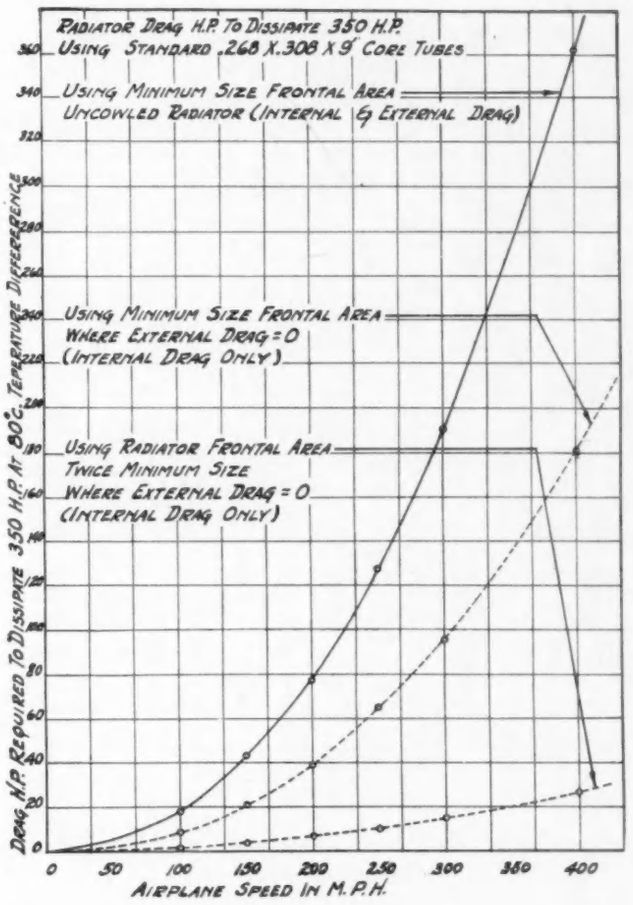
RADIATOR DRAG H.P. TO DISSIPATE 350 H.P.
USING STANDARD .268 X.308 X 9" CORE TUBES
USING MINIMUM SIZE FRONTAL AREA
UNCOWLED RADIATOR (INTERNAL & EXTERNAL DRAG)
USING MINIMUM SIZE FRONTAL AREA
WHERE EXTERNAL DRAG = 0
(INTERNAL DRAG ONLY)
USING RADIATOR FRONTAL AREA
TWICE MINIMUM SIZE
WHERE EXTERNAL DRAG = 0
(INTERNAL DRAG ONLY)
DRAG H.P. REQUIRED TO DISSIPATE 350 H.P. AT 80° C. TEMPERATURE DIFFERENCE
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360
0 50 100 150 200 250 300 350 400
AIRPLANE SPEED IN M.P.H.

Fig. 6 – Radiator Drag Horsepower Required to Dissipate 350 Hp. at Various Speeds

the power loss shown by this curve, the radiator frontal area is reduced to a minimum that will cool the engine. It is obvious that refinement and improvement in the efficiency of the radiator installation is essential to keep up with the general refinement that has cleaned up the entire airplane. The reason for the tremendous increase in the cooling drag with speed is the fact that cooling increases as the 0.83 power of the air flow whereas the power absorbed increases as the cube of the air flow.

The article in the August, 1935, issue of *Aviation*[2] presented the analysis of tests at the Materiel Division which indicated the approach to the radiator problem that will permit the refinement and reduction of radiator drag consistent with the higher airplane speeds and engine horsepowers.

The main basis of this analysis is the separation of the drag horsepower into two parts: First, the internal horsepower loss that results from the quantity of air flowing and the pressure drop through the radiator core, and second, the external power loss that results from the frontal area and the deflection of the approaching air around the radiator. With the analysis of the drag into these two parts, the internal power loss can be controlled to maintain the same air flow through the core at higher airplane velocities by restricting the air flow with cowling (using the large surface necessary for the slow velocity airplane at the higher speeds) and maintaining the internal drag of the slower airplane.

The external drag can be reduced by cowling that avoids turbulence and by locating the radiator within the essential structure of the airplane such as, for example, the wing. It is believed, and tests would indicate, that by this procedure the external drag eventually will approach zero.

The upper dotted line in Fig. 6 represents the drag horse-

[2] See *Aviation*, August, 1935, pp. 19-21; "Bigger and Better Radiators," by Weldon Worth.

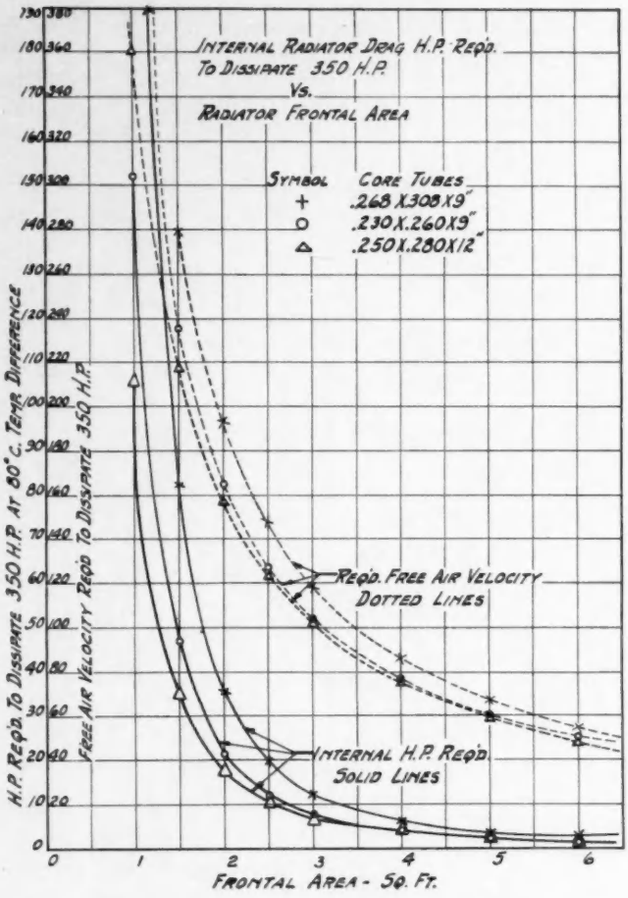
Internal Radiator Drag H.P. Req'd. To Dissipate 350 H.P. Vs. Radiator Frontal Area
Symbol Core Tubes
+ .268 X.308 X9"
○ .230 X.260 X9"
△ .250 X.280 X12"
Req'd. Free Air Velocity Dotted Lines
Internal H.P. Req'd. Solid Lines
H.P. Req'd. To Dissipate 350 H.P. At 80° C. Temp. Difference
Free Air Velocity Req'd To Dissipate 350 H.P.
Frontal Area - Sq. Ft.

Fig. 7 – Variation in Internal Drag Horsepower with Frontal Area for Three Radiator Cores

power of the radiator with minimum frontal area if the external drag were actually zero.

The lower dotted line shows the power loss that would be obtained at the same speeds if a radiator twice as large as the minimum size were used and the air flow properly restricted by cowling without turbulence. The weight horsepower is not considered in this curve.

The difference between 92 and 15 hp. (in Fig. 6 at 300 m.p.h.) represents the saving in drag that is possible at higher speeds by proper design, but this saving cannot be achieved unless the air flow is restricted properly by cowling and a large radiator is used with low air velocities past the cooling surface.

This comparison also indicates the inefficiency of a retractable radiator that uses the smallest possible extension of radiator as compared with using all the radiator with controlled cooling.

This discussion illustrates the need for some method of designing a radiator installation that gives consideration to the essential factors and, even though all of the answers are not available, a usable method for designing radiators and radiator illustrations will be presented.

Design Method

If a person upon being introduced to an octopus knew at once which tentacle to shake, such a person should know which angle of the radiator problem to consider first. But the radiator location is as good a starting place as any, and it certainly should be determined at the same time when the engine, fuselage, and wing combination are selected. When the radiator is an afterthought, it always will be that kind of an installation.

There are a great number of possible radiator locations, but the most promising locations are within the essential airplane structure where the radiator offers no projections into the free air stream. When the radiator is located internally, additional radiator area will reduce not only the internal drag but will also reduce the external drag and make the curves just shown more nearly applicable. As will be explained later, the increased radiator size makes it possible to increase the exit velocity of the cooling air and this reduction in the differential velocity between the cooling air and the free air accounts for the decrease in external turbulence. Some radiator installation tests have been conducted and, although they are confidential at this time, it can be stated that the tests indicate radiator locations within the wing are efficient.

In selecting the radiator location, attention should be given to the possible size of the radiator because this determination of size is the next problem.

Fig. 7 shows the variation in internal horsepower with frontal area for three radiator cores. Since this figure represents the actual energy lost in forcing the air over the cooling surfaces, it represents the drag values that can be approached by eliminating external drag. The curves are based upon a heat dissipation of 350 hp., which is a reasonable figure for a 1000-hp. engine. These values show a cooling drag of about ½ per cent of the brake horsepower with 3½ sq. ft. of radiator frontal area. The required free air velocity to cool with the various frontal areas is shown by the dotted lines. This required velocity can be converted readily to the necessary air-pressure drop across the core by referring to Fig. 8. For example, from Fig. 7, to cool with 3½ sq. ft. of 0.230 x 0.260 x 9 in. core tubes requires a free air velocity of 91 m.p.h. From curves in Fig. 8 this velocity requires approximately 3300

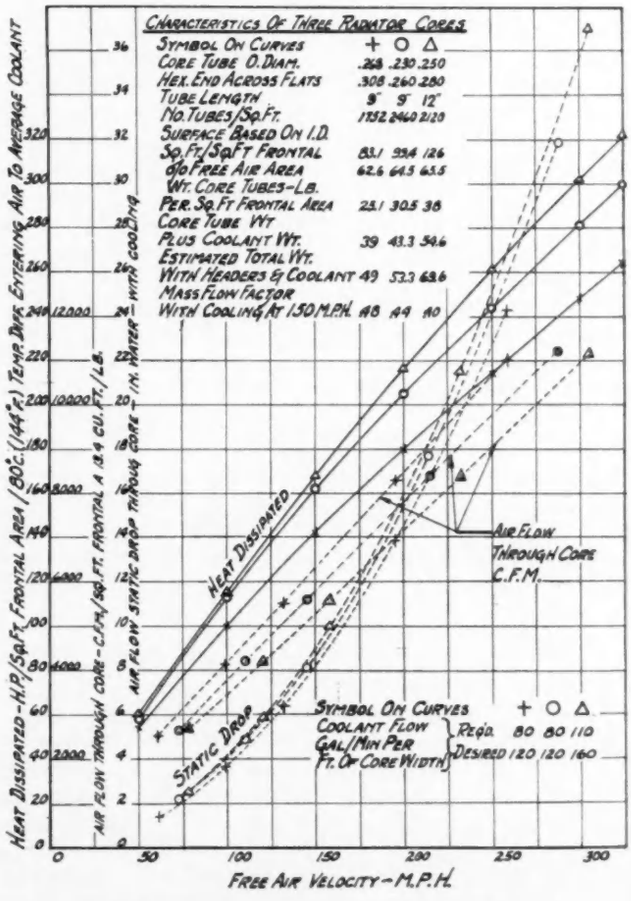
Characteristics Of Three Radiator Cores
Symbol On Curves + ○ △
Core Tube O. Diam. .268 .230 .250
Hex. End Across Flats .308 .260 .280
Tube Length 9" 9" 12"
No. Tubes/Sq. Ft. 1752 2440 2120
Surface Based On I.D.
Sq. Ft./Sq. Ft. Frontal 83.1 93.4 126
% Free Air Area 62.6 64.5 65.5
Wt. Core Tubes-Lb. Per Sq. Ft. Frontal Area 25.1 30.5 38
Core Tube Wt. Plus Coolant Wt. 39 43.3 54.6
Estimated Total Wt. With Headers & Coolant 49 53.3 68.6
Mass Flow Factor With Cooling At 150 M.P.H. 48 49 110
Heat Dissipated
Air Flow Through Core C.F.M.
Static Drop
Symbol On Curves + ○ △
Coolant Flow Gal./Min. Per Ft. Of Core Width: Req'd. 80 80 110; Desired 120 120 160
Heat Dissipated - H.P./Sq. Ft. Frontal Area (80° C. (144° F.) Temp. Diff. Entering Air To Average Coolant
Air Flow Through Core - C.F.M./Sq. Ft. Frontal Area (3.4 Cu. Ft./Lb.
Air Flow Static Drop Through Core - In. Water With Cooling
Free Air Velocity - M.P.H.

Fig. 8 – Characteristics of Three Radiator Cores

cu. ft. per min. per sq. ft. at a static drop of approximately 3.4 in. of water. Fig. 9 is based upon Fig. 7 with estimated values of horsepower loss due to the weight of the radiator added to the internal drag. This increases the cooling drag from ½ per cent to about 3 per cent of the brake horsepower.

Without an efficient internal location a value for external drag must be added to this figure. Some wind-tunnel tests at the Materiel Division woud indicate that, if the radiator were located to avoid interference drag, the external drag for such a radiator at 300 m.p.h. could be estimated at approximately 36 hp. This addition brings the total radiator drag to about 6½ per cent of the brake horsepower. If this value is used for comparison, it should be remembered that it includes the radiator weight, and also that it is hoped to reduce the 3 per cent external drag to a much lower value. The other drag probably will not be reduced appreciably.

The problem of selecting the optimum core size may be accomplished if desired by evaluating the total drag horsepower in the following equation:

The total power = internal power + power due to weight + power due to frontal area

$$\text{hp.} = Ak w^3 + \tfrac{1}{4}\,\frac{AWV}{375} + F \times 0.0032\,\frac{AV^3}{375}$$

$$A = \frac{300}{cw^{0.83}}$$

Where A = Frontal area required to cool 300 hp. at 100 deg. fahr. temperature difference

k = A constant of internal hp. from relation Internal hp./sq. ft. frontal = kw^3

w = Air flow in lb./sec./ft.² frontal area at standard condition

W = Radiator weight in lb./ft.² frontal area

F = Factor is equal to ratio of:

$$\frac{\text{External drag 1 ft.}^2\text{ frontal area}}{\text{Drag of 1 sq. ft. flat plate}}$$

V = Free air velocity in m.p.h. (Assumed same as airplane speed in this case)

c = Heat dissipation constant for relation. Hp. dissipated = $cw^{0.83}$

L/D of airplane is taken as 4. (Probably low, but low figure is prejudicial to conclusions)

Substituting for A:

$$\text{hp.} = \frac{300\,k}{c}\,w^{2.17} + \frac{300\,WV}{375 \times 4\,cw^{0.83}} + \frac{0.0032 \times 300}{375}\,\frac{V^3F}{cw^{0.83}}$$

Taking the derivative, equating it to zero, and solving for condition of minimum horsepower:

$$w = \sqrt[3]{0.000255\frac{WV}{k} + 0.00000326\frac{V^3F}{k}} \qquad (1)$$

and simplifying the previous equation for horsepower:

$$\text{hp.} = 300\,\frac{k}{c}\,w^{2.17} + 0.2\,\frac{WV}{cw^{0.83}} + 0.00256\,\frac{V^3F}{cw^{0.83}} \qquad (2)$$

With this equation one may calculate very accurately the best size, based upon very approximate assumptions and, although the theoretical variation should be kept in mind, the curves already referred to illustrate the nature of this variation. It is believed that, in most cases, the actual selection can be based upon these curves since the size probably will be determined to a large extent by practical limitations

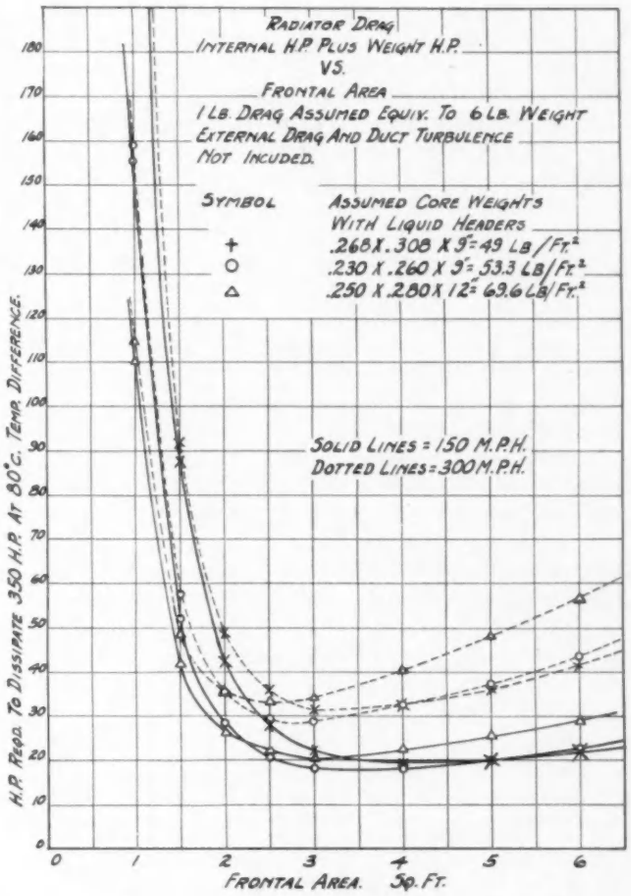

Fig. 9 – Radiator Drag (Internal Horsepower plus Weight Horsepower) Vs. Frontal Area

such as the space between two spars or the size of an oil tank. The important point evinced by these curves is the fact that the radiator size is not critical within certain limits so long as ample area is used. Within the variations of these limits the benefits of a large size and low drag compensate for the increased weight. It also provides a very valuable safety factor whereby more cooling can be obtained if desired by a cowl change.

In general, the structural problems of the strength of headers and sidewalls, as well as of weight, favor the use of 9 in. cores.

Having selected the location, the core, and the radiator size, one should then proceed to calculate the air flow necessary to give the desired cooling. This step coordinates data from varied sources – the engine heat rejection, the radiator heat dissipation capacity, the air-flow characteristics of the airplane, and the radiator location. If this step is accomplished without serious error, it means that all the little B.T.U.'s in several laboratories have been well behaved and properly supervised. The curves in Fig. 8 show the radiator characteristics that may be used to calculate the air flow necessary.

It is admitted that there is a vast abyss in available data on the correlation of these steps but, as we get more installations with better tests, the reliability of design practice will advance. The amount of cooling by direct air flow in the engine compartment is one of the elusive quantities, and at present it is good practice to use not only the design method, but the "Jake-did-this-way" method for checking purposes.

Having estimated the necessary air flow, the next step is the design of the air entrances and exits. A sharp-edged, smooth, gradually tapered entrance is, under test conditions,

very good if the area is exactly the size that will admit the necessary cooling air through the entrance without any change in velocity until it is inside the air duct. Although it is efficient if exactly correct, it is quite critical and, when too large or too small, the air flowing at an angle over the sharp edge will introduce considerable turbulence and drag. To avoid this critical aspect, the entrance should be rounded after the style of air-cooled engine cowls but the minimum entrance should never require an increase in the free air velocity at the entrance, but it may be slightly larger than necessary if the air spilling over is well directed, especially if there is not room for a proper expansion.

The expansion and smoothness of the air flow inside the entrance duct are extremely important since, for efficiency, it must convert the velocity head into pressure head. The entrance, of course, should be straight, smooth, free from turns, and the total diverging angle of the sides of the duct should always be less than 15 deg. Naturally this design can never be done in an airplane, but again the ideal should be kept in mind as it is disregarded. It always helps in maintaining that critical pessimistic attitude of a cautious engineer.

When designing for the efficiency of an over-size radiator, the exit duct exercises the control of the air flow. It is important to exercise the air flow control with the exit duct and not the entrance. The entrance should merely be designed for the air flow that is established by the exit duct and the core resistance. To illustrate the difference in power lost by these two methods restricting air flow, the curves in Fig. 10 show a comparison of the internal power. The solid curve is an interesting characteristic of the internal power variation with core restriction. Regardless of the kind of restriction or type of core, so long as there is no cowl restriction, the internal power starts at zero where there is no core resistance. It increases to a maximum at a mass flow factor of 0.58 and decreases to zero as the air flow is decreased to zero by the core resistance. Any turbulence, shutters, or restriction to air flow that does not increase the exit velocity follows this curve. With any given core resistance, if air flow is restricted by increasing the exit velocity and conserving the kinetic energy of the air, the internal power will decrease in the order indicated by the dotted line. For example, at 300 m.p.h. with a mass flow factor of 0.4 (that of a 12-in. core tube) the unrestricted internal power loss is 61 hp. If the flow is restricted to half that value by shutters at the core, the internal horsepower would be 34.5 hp. but, if it were restricted by increasing the exit velocity, the internal horsepower would be 7.5 hp. – a saving of 27 hp. The importance of this saving in controlling air flow between level flight and climb or summer and winter decreases with a reduction in the maximum internal radiator drag.

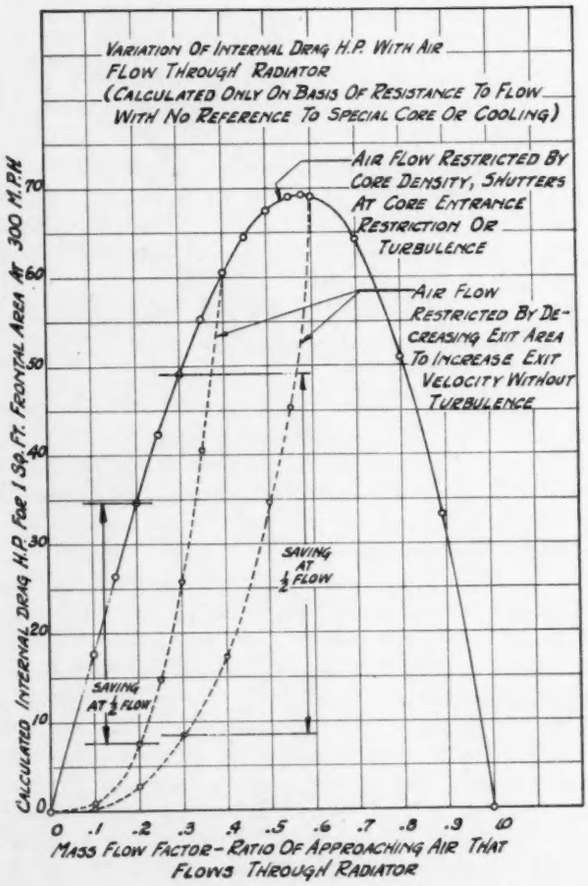
VARIATION OF INTERNAL DRAG H.P. WITH AIR FLOW THROUGH RADIATOR
(CALCULATED ONLY ON BASIS OF RESISTANCE TO FLOW WITH NO REFERENCE TO SPECIAL CORE OR COOLING)
AIR FLOW RESTRICTED BY CORE DENSITY, SHUTTERS AT CORE ENTRANCE RESTRICTION OR TURBULENCE
AIR FLOW RESTRICTED BY DECREASING EXIT AREA TO INCREASE EXIT VELOCITY WITHOUT TURBULENCE
SAVING AT ½ FLOW
SAVING AT ½ FLOW
CALCULATED INTERNAL DRAG H.P. FOR 1 SQ. FT. FRONTAL AREA AT 300 M.P.H.
0
10
20
30
40
50
60
70
0 .1 .2 .3 .4 .5 .6 .7 .8 .9 1.0
MASS FLOW FACTOR – RATIO OF APPROACHING AIR THAT FLOWS THROUGH RADIATOR

Fig. 10 – Variation of Internal Drag Horsepower with Air Flow Through Radiator

The internal drag of a radiator can be evaluated by considering the flow of a single particle through the duct. It enters the duct with the full velocity head of the airplane and no static head. As it slows down and expands to the area of the core passage, the velocity head is largely converted into pressure head. Part of this head is lost in pressure drop through the core, and the remainder is reconverted to velocity head at the radiator duct exit. It leaves the exit duct with substantially zero static head, and a velocity head less than its entrance velocity head by the amount necessary to account for the head loss of the duct and of the core.

If it should leave with the same velocity as it enters the duct, it would mean there was no intervening power loss, no internal radiator drag, and hence no radiator. It is evident that the exit area should be restricted as much as possible to provide as high an exit velocity as can be maintained and still account for the necessary losses because the decrease in velocity from the entrance to the exit represents the energy that has been lost. If the flow is not controlled properly, in spite of a large radiator that would otherwise permit a high exit velocity and low drag, the low drag will not be obtained and the difference will be lost by some form of turbulence in the duct. This turbulence may be introduced by a restriction at the entrance if there is an air flow restriction at this point. This same problem exists with air flow through an air-cooled engine.

The actual effect of altitude on the cooling problem is quite a surprise to most people. With a fully supercharged engine it is no more difficult to meet the Prestone and oil-cooling problems at 30,000 ft. than it is at sea level with the same power output. The reason for the common opinion to the contrary is the water-cooled engines which had to lower the temperature with increasing altitude to avoid boiling. The normal boiling point of Prestone is not reached until an altitude over 30,000 ft. is attained.

The decrease in density with altitude decreases the cooling in direct proportion but the decrease of about 2 deg. cent. per thousand feet altitude in air temperature will compensate for the density if the original temperature difference is not too high. Air-cooled engines probably are more difficult to cool at altitude because of the higher temperature.

To illustrate the design process a sample calculation will be carried through from the beginning. The assumed problem will be a fully supercharged 1000-hp. engine with a heat rejection of 350 hp. and a coolant flow of 200 gal. per min. An internal installation will be used with ample room for a radiator of 3 or 4 sq. ft. The speed of the airplane at maximum

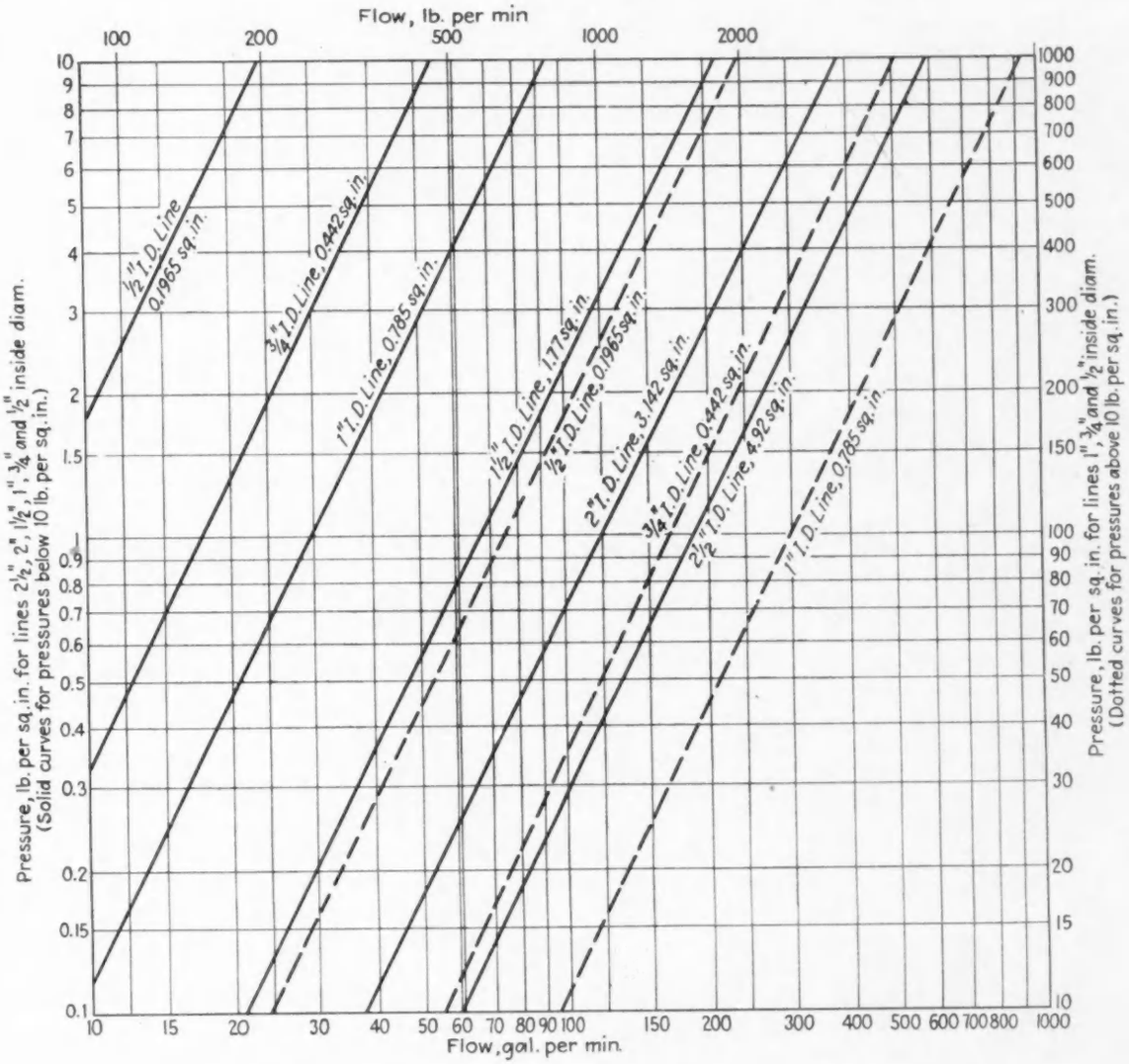
Flow, lb. per min
100
200
500
1000
2000
½" I.D. Line 0.1965 sq. in.
¾" I.D. Line, 0.442 sq. in.
1" I.D. Line, 0.785 sq. in.
1½" I.D. Line, 1.77 sq. in.
1½" I.D. Line, 0.1965 sq. in.
2" I.D. Line, 3.142 sq. in.
¾" I.D. Line 0.442 sq. in.
2½" I.D. Line, 4.92 sq. in.
1" I.D. Line, 0.785 sq. in.
Pressure, lb. per sq. in. for lines 2½", 2", 1½", 1", ¾" and ½" inside diam. (Solid curves for pressures below 10 lb. per sq. in.)
Pressure, lb. per sq. in. for lines 1", ¾" and ½" inside diam. (Dotted curves for pressures above 10 lb. per sq. in.)
Flow, gal. per min.

Fig. 11 – Velocity Pressure Head Vs. Water Flow for Lines or Headers of Various Areas

climb will be 130 m.p.h., and it will have a top speed of 300 m.p.h.

Looking at Figs. 7 and 9 and considering the various aspects, a selection of 3.5 sq. ft. of frontal area is made using 0.230 × 0.260 core tubes 9 in. long.

The selection of the 3.5 sq. ft. area instead of a smaller value requires less air flow, provides for efficient increase of the cooling if desired, and it is probable that the air duct losses not estimated in the curves would make the 3.5 sq. ft. area an optimum. If an external radiator were used with frontal area drag, a smaller radiator would be preferable. In this installation, Fig. 9 indicates that as low as 2.5 sq. ft. would be reasonably efficient but from Fig. 7 this area requires a free air velocity of 130 m.p.h. available at the core whereas the 3.5 sq. ft. area requires only 89 m.p.h. With a climb of 130 m.p.h., the need for a margin of safety favors the larger size.

To dissipate the 350 hp. the air flow and equivalent velocity can also be taken directly from Fig. 8, reducing the 350 hp. to 100 hp. per sq. ft. frontal area (The temperature difference is the difference between the maximum summer temperature and the allowable engine coolant temperature) and multiplying the air flow (3250 cu. ft. per min.) by 3.5 to give the total air flow required, namely, 11,400 cu. ft. per min. The static drop as indicated by Fig. 8 for this air flow is 3.4 in. H_2O.

These curves are based upon a density of 13.4 cu. ft. per lb., the density at 68 deg. fahr. and a barometer of 29.92 in hg. At sea-level density with the maximum assumed air temperature for cooling of 100 deg fahr., the specific volume is 6 per cent more, or 14.2 cu. ft. per lb. Therefore, the air flow and pressure drop for this density should be increased by direct proportion to 3.6 in. H_2O drop and a total flow of 12,100 cu. ft. per min.

Assuming entrance and exit duct losses of 30 per cent of the maximum entrance velocity head, 130 m.p.h., the loss will be 30 per cent of 8.4 in. H_2O which equals 2.5 in. H_2O. This value gives a total duct and core loss of 2.5 plus 3.6, or 6.1 in. H_2O. With 8.4 in. H_2O available at the entrance there is 8.4 minus 6.1, or 2.3 in. H_2O available which permits an exit velocity of 68 m.p.h. With 12,100 cu. ft. per min. flowing the required entrance area at 130 m.p.h. is 1.06 sq. ft., and at 68 m.p.h. the exit area is 2.01 sq. ft. Tests are not available to show whether this exit area needs an addition or correction for the decrease in density resulting from heating the air with the radiator. If such a correction is needed, it would not exceed 8 per cent and probably be much less.

No entirely satisfactory shutter design has been tested by the Division but, for maximum efficiency, it should restrict the exit area without turbulence. At a high-speed velocity of 300 m.p.h. with something like 44.5 in. H_2O available, the

exit velocity, corresponding to 44.5 minus 6.1, or 38.4 in. H_2O, is 279 m.p.h. Thus the area could be 0.5 sq. ft. for maximum efficiency. The entrance could, of course, be restricted at high speed but it is probably more practical to use a well-rounded entry with a fixed area.

When heat rejection for a radiator is plotted against liquid flow, it is practically constant at higher flows but falls off rapidly below certain values. All data have been based upon high liquid velocities, and care should be taken to maintain the velocities listed in Fig. 8. Velocities in excess of these values are objectionable only because of the unnecessarily high pressure loss. The objections to higher pressure are chiefly the pump horsepower, and the additional pressure on the lines, hose connections, and the radiator. Its effect upon the flow also may be material, depending upon the pump characteristics, but radiator restriction does not break down the circulation in the Materiel Division cooling system as it would in an automobile-type system. In the design example with 200 gal. per min. flow, the width of the flow area preferably would be about 20 in. Thus radiator frontal dimensions 20 in. wide and 25.2 in. high, with the flow of both banks entering at the top and flowing downward to the exit at the bottom, would be desirable.

The quantity of liquid flow through liquid-cooled engines is greater than one visualizes, and great care must be used to avoid restrictions, turbulence, and pressure loss in the radiator headers, inlets, and outlets. The velocity and the associated velocity head is the best index of the extent of this problem. To aid in approximating this value, the logarithmic chart, Fig. 11, conveniently shows this value. Example: with 200 gal. per min., the velocity head in a 1-in. line, or in a header with an area of 0.785 sq. in., would be 45 lb. per sq. in. and the same line at a flow of 50 gal. per min. will have a velocity head of 2.8 lb. per sq. in.

In mounting the radiator it is advisable to avoid a cradle with a vise-like screw adjustment that inadvertently can be used to crush the radiator. Care also should be taken to distribute the support and to avoid concentration of the supporting forces at one point, especially with large radiators. The radiator cradle or support should be insulated against vibration from the airplane structure by rubber bushings that provide flexibility in all directions. A deflection under the radiator's own weight of approximately ⅛ in. is believed to be a desirable degree of flexibility, and the use of this flexibility, in some cases, has eliminated a condition of repeated radiator failures.

At the end of the first lap of a step-by-step process in designing a radiator installation, it probably is advisable to return again to the first factors that were considered to relate these factors to the design data of the later steps.

Radiator Drag Test

Inquisitive people frequently ask embarrassing questions as to just how much the presence of a radiator actually slows down an airplane. The Materiel Division made one very interesting test to determine the answer to this question but, as often happens, the value of the data became lost in the rush of progress, and the airplane upon which the tests were made is now obsolete. Unless the trends can be anticipated by at least three or four years, tests of this kind are discouragingly useless. In this test the speed measurements were taken on a normal biplane pursuit-type airplane which had a radiator located between the flared landing-gear struts, as shown in Fig. 12 (*left*). It was not a good installation, but was typical of the period of 1930-1933. The radiator then was removed with smooth fairing, as shown in Fig. 12 (*right*) and a check speed run was made without the radiator. The novelty of the test lay in the method of engine cooling that was employed. The main gas tank was filled with water and the latent heat of vaporization of this water, with the aid of an internally located heat exchanger, provided admirable cooling of the Prestone for a flight of 15 to 20 min. duration, and gave the fascinating appearance of an aerial steam locomotive. In this case the actual speed was increased from 187.5 m.p.h. to 204 m.p.h., an increase of 16.5 m.p.h. by the removal of the radiator. In this case 22½ per cent of the engine horsepower was being used to drag the radiator which had 1.5 sq. ft. of frontal area and an equivalent flat-plate-area drag of 2.17 sq. ft. Obviously there was much interference drag.

Other tests on more modern airplanes with obsolete radiator installations have shown 12 per cent and 16 per cent of the power attributable to radiator drag, but tests indicate that, with an efficient radiator location, it will be possible to cool the engines with 4 to 6 per cent of the brake horsepower, possibly even less. This comparison will suggest the improvement that is possible in liquid-cooled-engine installations today as compared with those of a few short years ago.

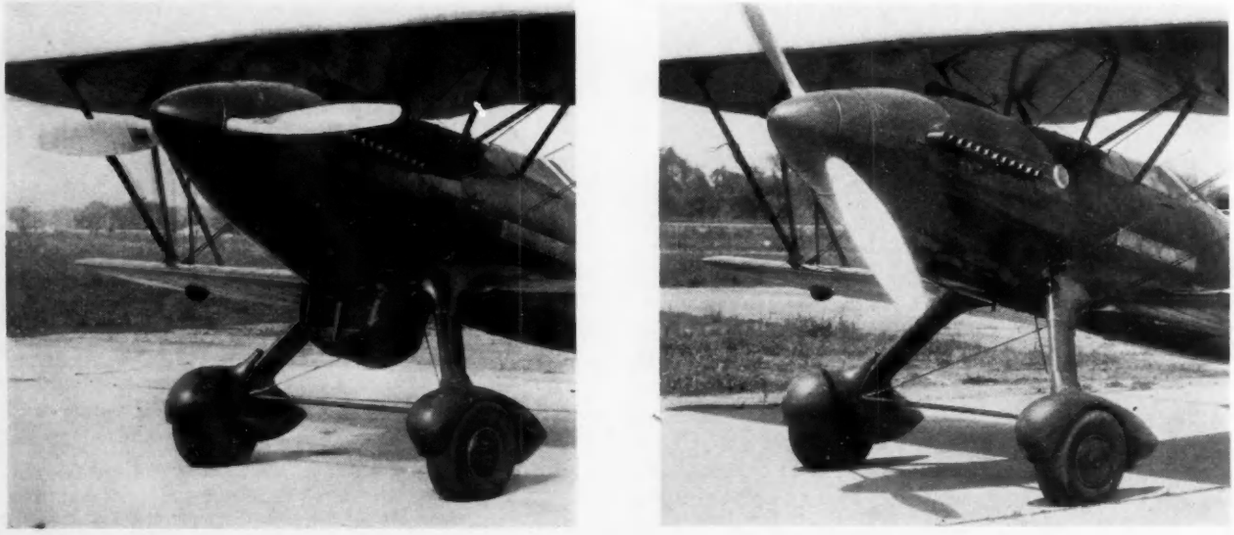

Fig. 12 – Pursuit-Type Biplane with Radiator (*left*) and without Radiator (*right*)